

DOMESTIC GRAYWATER HEAT RECOVERY FROM THE UIUC 2009 SOLAR
DECATHLON HOUSE

BY

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THESIS

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ABSTRACT

This study investigates the potential for waste water heat recovery in a residential application. A heat exchanger is utilized to transfer heat from the waste water drain line of a residence to the incoming water supply line. An experiment is conducted on the University of Illinois 2009 Solar Decathlon competition home to quantify the benefit of a heat recovery system that utilizes a counter flow flat plate heat exchanger in a balanced flow configuration. This system recovers heat in a non-regenerative scheme (meaning that it does not retain the discharge water for batch heat recovery) and is capable of reducing the hot water energy demand the home. This system is designed to benefit appliances like showers and sinks that drain simultaneous to the make-up water demand. The results of the experiment indicate that up to 37% of the thermal energy used by a shower can be recovered with the system developed in this study. These results are based on a drainage water temperature of 101 °F (38 °C) and a supply water temperature of 76 °F (24 °C) at an average effluent mass flow rate of 6.74 kg/s. Flow rate is determined to have a positive correlation to the magnitude of recoverable heat, although the heat recovery percentage has marginal difference for the low, medium, and high flow rates tested. The heat recovery percentage was 34%, 37%, and 33% for the low, medium, and high flow rates tested. It is further estimated that the heat recovery percentage would increase as the supply water temperature decreases. An estimate is conducted that shows the heat recovery percentage for each flow case would increase to 41%, 44%, and 38% respectively if the supply water temperature was reduced to 55 °F (12.8 °C).

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NOMENCLATURE

\dot{m}	Mass flow rate
\dot{Q}	Heat transfer rate
μ	Dynamic viscosity
a	Corrugation amplitude
A_o	Overall heat exchanger area
b	Corrugation depth
C	Heat capacity
COP	Coefficient of performance
c_p	Specific heat capacity
C_r	Heat capacity ratio
d_e	Equivalent diameter
d_h	Hydraulic diameter
DOE	Department of Energy
f	Moody friction factor
GFX	Gravity-film heat exchanger
H	Enthalpy
h	Convective heat transfer coefficient
HVAC	Heating, ventilation, and air conditioning
k	Thermal conductivity
L_p	Length from PHE port center-to-center
N	Number of channels
N_{plates}	Number of PHE plates
NTU	Number of transfer units
Nu	Nusselt number
PEX	Cross-linked polyethylene
PHE	Plate heat exchanger
Pr	Prandtl number
PVC	Polyvinyl chloride
Q	Volumetric flow rate
R_c	Heat capacity ratio
Re_h	Reynolds number based on D_h
T	Temperature
t	Heat exchanger plate thickness
u	Velocity
U_o	Overall heat transfer coefficient
w	Heat exchanger width
β	Corrugation angle
ΔT_{lm}	Log mean temperature difference
Λ	Corrugation pitch
ρ	Density
Φ	Area enlargement ratio
χ	Corrugation parameter
ε	Heat exchanger effectiveness

SUBSCRIPTS

h	hot
c	cold
i	in
o	out
w	wall
min	minimum
max	maximum

CHAPTER 1: INTRODUCTION

In 2008, it was estimated that the residential sector consumed 22% of the total energy used in the United States, and of that, 19% was utilized for water heating purposes (EIA, 2008). Figures 1 and 2 respectively show the distribution of energy consumption by sector and in the residential sector alone. Thus, it can be concluded that more than 4% of the energy consumption in the United States is dedicated to water heating in the residential sector alone. Further, it is estimated that 350 billion kWh is lost annually through building water discharge lines with notable potential for heat recovery according to ORNL (2005).

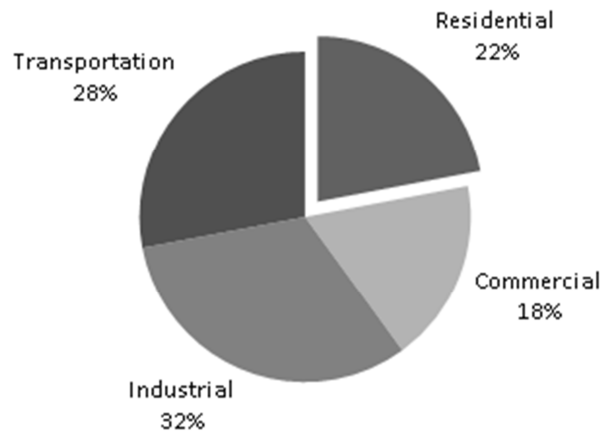


Figure 1. U.S. energy consumption by sector (EIA, 2008).

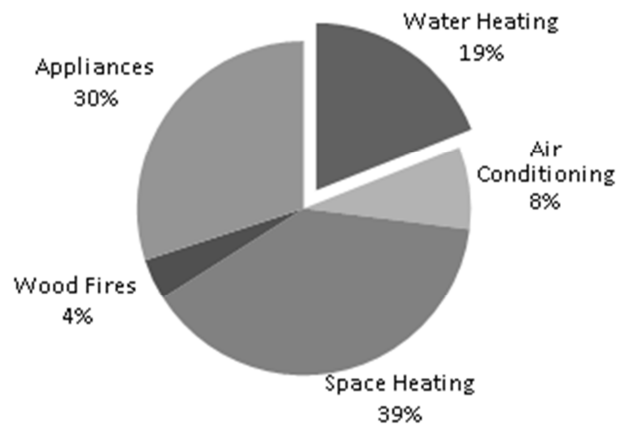


Figure 2. Energy use in the residential sector (EIA, 2008).

Overtime, there have been several studies conducted which utilize different methods for recovering the thermal energy remaining in residential discharge water. Among these methods used for recovering the low grade thermal energy are various types of heat exchanger, heat pump, heat pipe, and thermosyphon systems. Some systems are designed with thermal stores to hold recovered sensible heat for subsequent water demands, while some are designed for heat recovery concurrent to water demand. The placement of a heat recovery system within the plumbing structure of a building also varies between studies found in scientific literature. Also, there are multiple ways in which the recovered heat from residential discharge lines can be utilized. Each method and system has unique characteristics. The varying configurations of the different systems and methods are merely unique approaches to achieve the same fundamental operation of discharge water waste heat recovery; however certain concepts may align better with common engineering system design objectives and, therefore, prove to have greater potential over the others.

Among the common design objectives, functionality, efficiency, and reliability are considered when developing an engineered system (U.S. DOE, 2010). It is essential for a system design to function as it is intended and to be capable of completing the task for which it is created. But further, efficiency is important because it is a measure of how well the task is performed in comparison to the theoretical best performance. Reliability, however, should not be compromised at any point. Consideration should be given to the ability of the system to continue functioning amidst the adverse conditions which are encountered throughout lifetime operation. Therefore, reliability is an important parameter to consider due to implications on the practicality of the system. Another important factor that must be included in system design, especially when a consumer is intended to benefit from the system, is the overall cost.

Achieving the most cost effective design is a complex process of optimization for the most important parameters, such that, diminishing returns are avoided for performance enhancements.

As previously mentioned, the reliability of a system is a critical consideration when developing a system design at the conceptual level. The reliability of a system can be influenced by a number of design characteristics, each of which are often interrelated with functionality, efficiency, and cost effectiveness. However, a reliable system is often obtained by limiting unnecessary complexity when possible, as this practice will likely reduce maintenance requirements, but is generally accomplished at the cost of reduced efficiency. Therefore, it is important to justify design choices with reason, which in a capitalistic society is often driven by economics. A study of the existing system ideas will help identify performance characteristics to aid in determining the most cost-effective solutions to the residential waste heat recovery concept.

Aside from the methods utilized to recover residential waste water heat, there are options for how the heat can be utilized within a home after recovery thus contributing to the numerous system options. There are several influential factors that help dictate what type of recovery system is most appropriate for a given situation. One important factor to consider is the quantity and quality of thermal energy available, as well as the frequency distribution which the resource is available for recuperation. From an energy standpoint, it is logical to apply the recovered energy to whichever demand can benefit the most from the added input throughout the duration of the lifetime of the system. However, this may not always be the best option from an economic standpoint, as the cost of energy differs by source type and installation requirements vary. Using the heat recovery to supplement the operation of an appliance with a higher instantaneous operating cost (dollars per end use energy) for fewer hours of the year, may prove more

beneficial than supplementing a system with a lower instantaneous operating cost for more hours of the year.

Another factor to consider when choosing a system design is the construction state of the home. Integrating a heat recovery system with other residential systems while a house is unfinished will allow more selection and freedom to pursue the most ideal system in terms of energy conservation with less impact on installation cost. A home being retrofitted with a heat recovery system may have to suffice with a less ideal system to prevent substantial installation cost which would offset the economic benefit of heat recovery.

The climate in which a residence is located should also be considered when determining the design of the recovery system. A home that has more heating degree days may benefit from a heat recovery system that supports the space heating function of a house. Alternatively, a home with more cooling degree days may find more cost effectiveness for this concept when it is employed to help support the water heating function. Other existing factors such as, appliance choices, occupancy, lifestyle, building type, building layout, and building construction, can dictate the best residential waste water heat recovery system to employ. Each residence should be considered on a case by case basis if there is a desire to achieve the most efficient method for utilizing the recovered heat. However, universal system options may prove favorable for numerous residences that share only limited similarities if the most important factors are identified and addressed.

The goal of this study is to investigate heat recovery from domestic waste water and to help identify the predominant influential factors to system design and performance. The objectives include:

1. Identify residential waste water heat recovery system designs and characterize the potential benefits and limitations of each.
2. Design an experiment to validate the energy savings potential of waste water heat recovery published in scientific literature.
3. Provide data to help identify essential parameters for inclusion in future development of waste water heat recovery prediction models.

CHAPTER 2: REVIEW OF LITERATURE

2.1 HOT WATER CONSUMPTION CHARACTERISTICS

As previously mentioned, 4% of the total annual energy consumption in the U.S. is used for water heating in the residential sector. This figure is similar to reports issued by sources in Europe and Hong Kong. For 2006, it is estimated that 26% of the final energy use by sector is dedicated to households in the EU-27 (EEA, 2010). For 2007, a prediction is made that 14% of the energy consumed by households is used for water heating (ADEME, 2010). Assuming that no significant changes occurred within a year regarding the relative energy consumption of the household sector to the total energy used in the EU-27 annually, it can be inferred that approximately 3.6% of the energy consumed in the EU-27 is dedicated to water heating. The Hong Kong Electrical and Mechanical Service Department (2008) describes that in 2005, 19% of the energy consumption in Hong Kong is used in residential buildings and that 20% of this energy is dedicated to hot water production. From these statistics, it can be deduced that Hong Kong has a similar hot water energy demand compared to the United States. Approximately 3.8% of the total energy consumption in Hong Kong is used for heating water in residences. Hot water consumption is a significant energy expense for developed countries and warrants research into the conservation effort on this front.

There are numerous uses for hot water in a home or apartment. Hot water is commonly used by inhabitants for showering and bathing while appliances such as a dishwasher and clothes washer utilize the resource to aid cleaning processes. Hot water is also commonly administered to sinks for which it can serve a variety of purposes. With these common end-uses for hot water, there exists significant variability in the hot water consumption within a residence in regards to the quantity and quality used over a given duration as well as the frequency of use within the

same time. An accurate prediction of hot water consumption is a complex task that requires taking several factors into consideration.

It is important to understand the water consumption characteristics of a residence to help design a heat recovery system to the available waste water resource. Smith (1975) makes an early note that lifestyle and social class have a strong influence on the amount of energy a home expends on water heating. Other factors that influence the resource will be appliances with varying flow demands and temperature demands. Appliances like a shower or sink have the characteristic of draining simultaneous to operation as opposed to appliances like baths, dishwashers, and clothes washers that will accumulate and store hot water. The appliances that accumulate water each may have different capacities and operating temperatures. Another factor that affects water consumption is the number of occupants in a residence. A simple assumption is a greater occupancy will require more hot water consumption. This consumption will vary drastically with habits as preluded by Smith (1975). Understanding what factors influence the waste water resource of a home help engineers to develop residential heat recovery systems that are best suited to the varying conditions.

2.2 GENERAL HEAT RECOVERY SCHEMES

With energy conservation as an objective, one can employ numerous methods by which a residence can benefit from the recovery of thermal energy from the waste water discharge line. Any recovery of thermal energy from the drainage system is a beneficial gain for the home, as any heated water in the drainage system would normally be exhausted to the environment where it serves little to no purpose. There are several uses for the recovered energy, yet there is not one method that is the ideal for every situation. As previously mentioned, it is important to understand several characteristics about a home or apartment in order to select the heat recovery

method that can be the most beneficial for the inhabitants. Aside from the method of heat recovery, there are several system configurations that have been found in literature, each with unique features that help define how the system is most effectively utilized.

2.2.1 Regenerative System

A regenerative system is, henceforth, a label used to describe a heat recovery system that stores thermal energy recovered from the waste water discharge of a home to be utilized at a later time (ORNL, 2005). This type of system is beneficial for instances when hot discharge water does not coincide with the demand of the intended sink for the recovered thermal energy. When preheating incoming feed water with the recovered thermal energy, any case where a significant amount of hot water accumulates before being discharged serves to benefit from a regenerative system. The hot water accumulation in the residence decouples the simultaneity of discharge flow and the resupply to the hot water heater, thereby risking that the waste water will drain without any heat recovery if the hot water tank has finished filling. Large volumetric water discharges in homes or apartments that are associated with hot water accumulation are typically from bath tubs, dishwashing machines, and clothes washing machines. Dwellings in which these appliances and fixtures dominate the hot water consumption are likely to benefit from a regenerative system.

An advantage of the regenerative system is that it is also capable of recovering the thermal energy from hot water demands that drain in a simultaneous manner to the incoming feed water supply. However, a regenerative system could prove overly complicated and less cost effective if there is a dominant presence of these type of fixtures compared to those which accumulate water and discharge out of synchronization with the make-up water demand. Another advantage that a regenerative system has is the ability to recover thermal energy from

large volumetric waste discharges over a longer period of time, which reduces the necessary overall heat transfer coefficient of a heat recovery device to achieve high recovery effectiveness.

2.2.2 Non-Regenerative System

A non-regenerative system is, henceforth, a label used to describe a heat recovery system that has no thermal storage. The waste water exchanges heat with the intended thermal sink while continuing relatively unimpeded to the waste water discharge of the home (ORNL, 2005). This type of system can be used when warm waste water generation coincides with the end-use demand for the recovered thermal energy. When using the recovered heat to preheat incoming feed water, this system performs favorably with fixtures that do not accumulate hot water, such as showers and sinks, where the demand for hot water coincides with the warm water waste discharge. Because there is no water storage, non-regenerative systems typically benefit from a reduced size in comparison to regenerative systems.

2.2.3 Distributed and Centralized Recovery

A distributed heat recovery system is, henceforth, the label used to describe a system that is located on the discharge line from a specific appliance, as opposed to centralized recovery, which is the label that will further be used to describe a scheme that is located on the single discharge line of all drainage sources. The distributed recovery system benefits from the ability to target specific drainage sources for heat recovery, which consequently eliminates the need to design for all waste water contents. An example from literature is the system developed by Wong et al. (2010) that uses only shower water drainage to preheat the cold water shower feed. This system eliminates the need to design a heat exchanger capable of handling the solid waste from toilets and garbage disposals. The distributed system also benefits from the close proximity to the drainage source, and therefore reduces the heat lost through drainage pipes to the

environment. Conversely, the centralized system is an appealing concept that avoids multiple heat recovery systems if more than one source is desired to target for heat recovery. This type of system does in fact need to be designed to handle drainage flow from all the fixtures and appliances that drain into the common line.

2.3 SYSTEM PERFORMANCE

2.3.1 Heat Pipe System

A heat pipe, as described in the following by Mills (1995) and pictured in figure 3, is an evaporator-condenser device which uses conduction, phase transition, and convection to transfer heat from one temperature region to a lower temperature region. Heat passes by conduction into a working fluid in the evaporator section within the sealed heat pipe, whereby, it is evaporated, causing a localized vapor pressure increase. This differential pressure moves the vapor to the other end of the pipe which is at a lower temperature, and therefore, has a lower vapor pressure (Zumdahl, 2003). Heat is transferred to the lower temperature region as the working fluid vapor is condensed into a wick structure which then enables the fluid to return to the evaporator through capillary pumping.

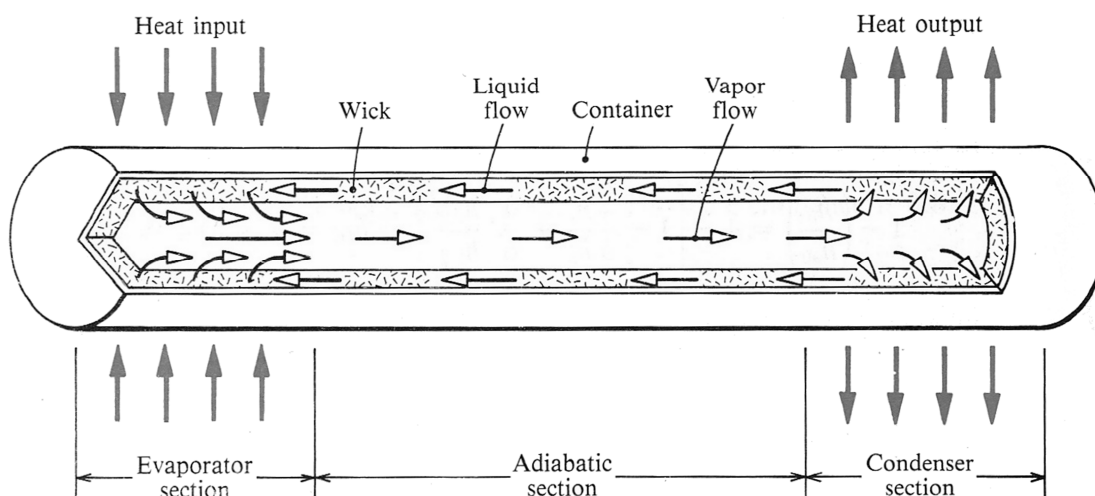


Figure 3. Diagram of a heat pipe (Mills, 1995).

A significant advantage of using a heat pipe is the typically higher heat flux rating compared to an equal sized solid copper pipe. This can be attributed to the process of heat transfer via latent heat rather than sensible heat and conduction. Given this function of heat pipes, a working fluid with a high enthalpy of vaporization is ideal because it allows a low internal vapor flow to carry significant amount of heat. Other benefits that heat pipes offer are the simple, passive design with no moving parts that require maintenance.

Behrmann et al. (1982) developed a waste water heat recovery system that employed a series of heat pipes, with water as the working fluid, which charged a stratified water tank with the remaining thermal energy from the waste discharge water stream. The study found that up to 90% of the usable thermal energy in the waste water stream could potentially be recaptured and used to heat the water in a stainless steel insulated storage tank. Figure 4 depicts a schematic of the heat pipe and storage tank system that was developed for this study. The heat pipes can be observed at an incline with respect to the horizontal, thereby allowing the heat pipes to work in a gravity-assisted mode. This, in turn, helps the working fluid return to the evaporator through the wick from the condenser. This configuration helps enable a diode-like behavior within the heat pipe, thereby enabling a dominant one way heat flow direction. This behavior, combined with the stratified tank design, allows water from the waste discharge stream to contribute heating to the appropriate level within the tank, while preventing the waste stream from being charged by higher temperature water in the supply tank that was previously heated by hotter discharge water.

The heat pipe system is likely most practical for centralized recovery schemes, given the size of the unit. A unique feature of this system is that while it meets the previously mentioned definition for a non-regenerative recovery system (i.e. the waste flow is relatively unimpeded while continuing to the discharge), it has thermal storage capability, as defined to be a

regenerative system that can provide recovered heat at a later time not concurrent to the waste discharge.

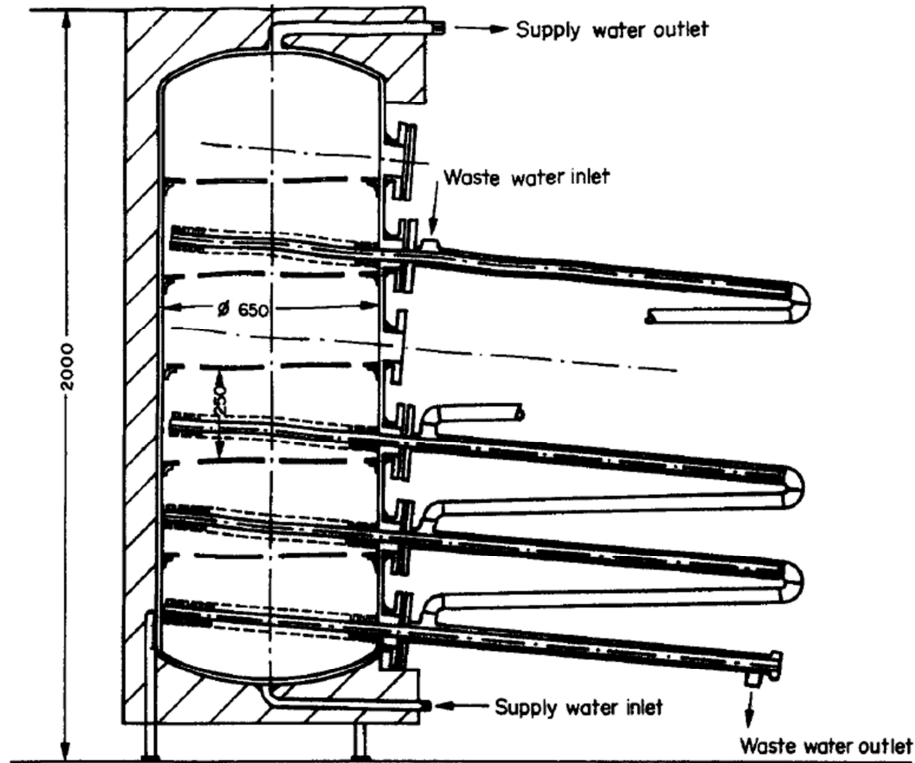


Figure 4. Schematic of heat pipe and storage tank system (Behrmann et al. 1982).

2.3.2 Thermosyphon System

One study completed by Kalema et al. (1985) investigated the potential value of using a thermosyphon to extract heat from the discharge line of a home to preheat fresh ventilation air. The study appears to have utilized a two-phase thermosyphon with a copper tube and fin evaporator immersed in a storage tank which collects discharge water from the home and overflows to the sewer. The condenser is located in the attic in the ventilation intake duct and is a copper tube covered with numerous radial aluminum pins. A two compartment storage tank with a partition is utilized to isolate and prevent incoming discharge flow from disrupting the temperature stratification in the other part of the storage tank which contains the evaporator.

Water enters the one side of the storage tank and can flow over the partition wall which has a small perforation at the bottom to permit circulation. It is suspected that this allows the cooling water in the evaporator side to push warm stratified water at the surface of the inlet tank side to the evaporator side over the partition wall via single phase thermosyphon action.

The two-phase thermosyphon heat transfer device operates on a principle as described by Naterer (2003), and pictured in figure 5, such that the working fluid in the evaporator section (located below the condenser) absorbs sensible and latent heat thereby driving vapor up to the condenser via buoyancy. In the condenser, the working fluid is condensed and releases latent heat causing the condensate to create a film on the interior of the of the thermosyphon and flow back down to the evaporator by gravitational force.

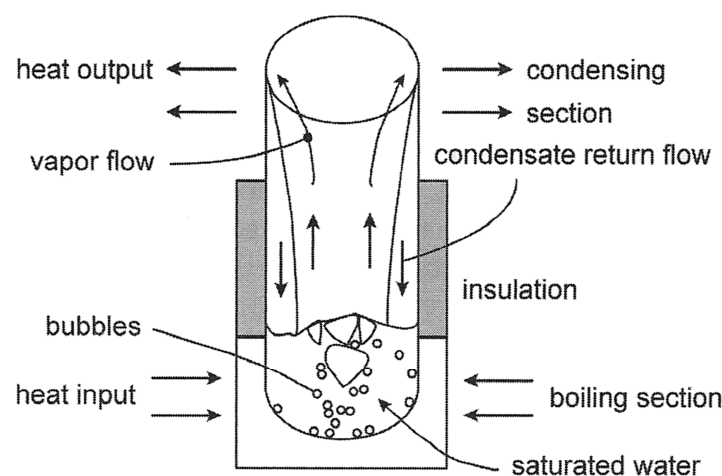


Figure 5. Diagram of a two-phase thermosyphon (Naterer, 2003).

The results obtained by Kalema et al. (1985) estimate that the thermosyphon was about 30% efficient during the heating season, which means that it recovered about 1/3 of the energy used to heat the tap water initially. It was noted that the efficiency of this system is strongly predominantly dependent on the outdoor air temperature because the storage tank water temperature did not experience significant variance. Therefore, it is concluded that the

efficiency of this system is lower than reported during the summer due to increased outdoor temperatures. It is also unlikely for a home to need heating in the summer, and therefore can render this system useless.

2.3.3 Heat Exchanger System

An early study completed by Smith (1975) investigated a unique batch recovery system that utilized a tank within a tank apparatus which can be seen in figure 6. Smith conducted a study on an establishment which was estimated to utilize over 85% of hot water for baths, dishwashing, and clothes washing.

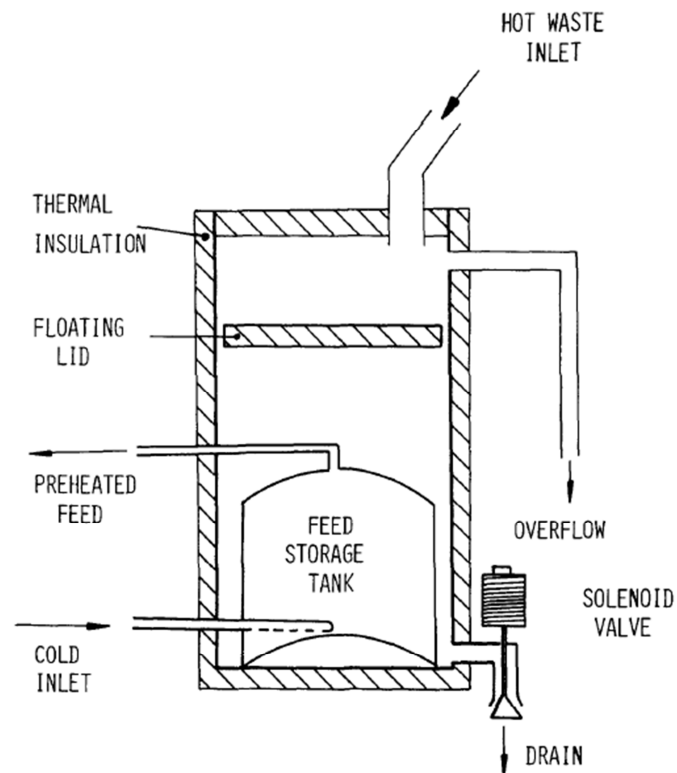


Figure 6. Tank in tank heat recovery system diagram (Smith, 1975).

Therefore, this particular study concluded that the development of a batch heat recovery system was beneficial, as opposed to employing a more traditional heat exchanger system which would need the capability to transfer greater than 100kW of thermal energy given the high rate of

discharge associated with a draining bath. The conventional heat exchanger option was also avoided due to the potential clogging from particles in the discharge stream and the reduced effect of heat exchange performance degradation from fouling compared with a system with a lower rate of heat transfer.

During the operation of this system, drainage water enters the waste tank from above. A feed storage tank is located at the bottom of the tank and stores cold supply water to be heated by the warm water in the waste tank before going to a boiler. If a temperature differential of about 3-5 °C is detected between the bases of the two tanks then a valve at the lower part of the waste tank closes to allow the warm water to accumulate. If the waste tank cools below the set temperature differential, then the drain valve opens and warm water that has stratified in the waste tank descends to the surroundings of the feed tank to permit further heat transfer. During a winter season, the system proved to save 31.7% on water heating energy. It was estimated that if the average annual feed temperature was taken at a few degrees higher, the energy savings would only reduce to approximately 30%. The increase in feed temperature reduces the temperature differential with the waste heat, and therefore, the recovery potential (Smith, 1975).

Another study, completed by Wong et al. (2010), investigated the potential for shower water heat recovery in high-rise buildings in Hong Kong. The system developed is a distributed scheme designed for localized heat recovery. The design uses a horizontal, single-pass, counter-flow heat exchanger below the shower drain to preheat the cold water feed to an instantaneous hot water heater that services the same shower. Figure 7 shows a diagram of the system design, while figure 8 depicts a schematic of the heat recovery device. This study predicts that with the given heat exchanger design of 1.5 meters in length and 50 mm in diameter, it could support a 4-15% savings on hot water heating energy.

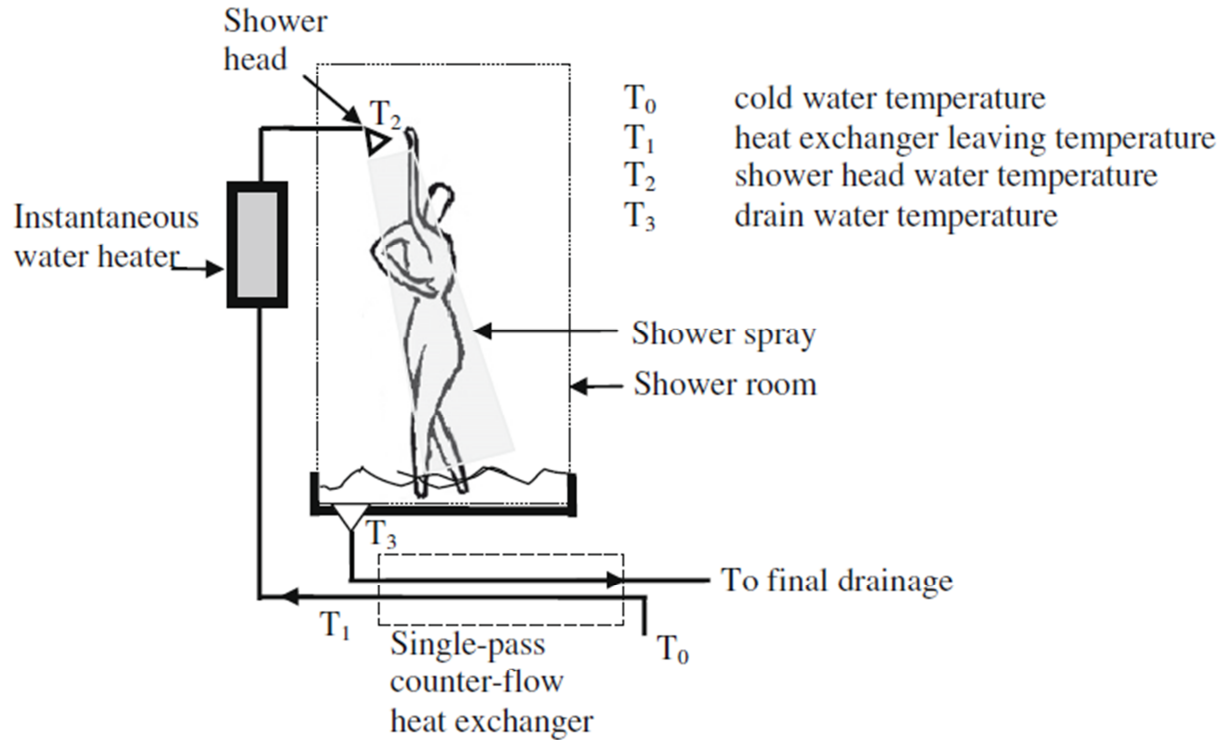


Figure 7. Distributed heat recovery system (Wong et al., 2010).

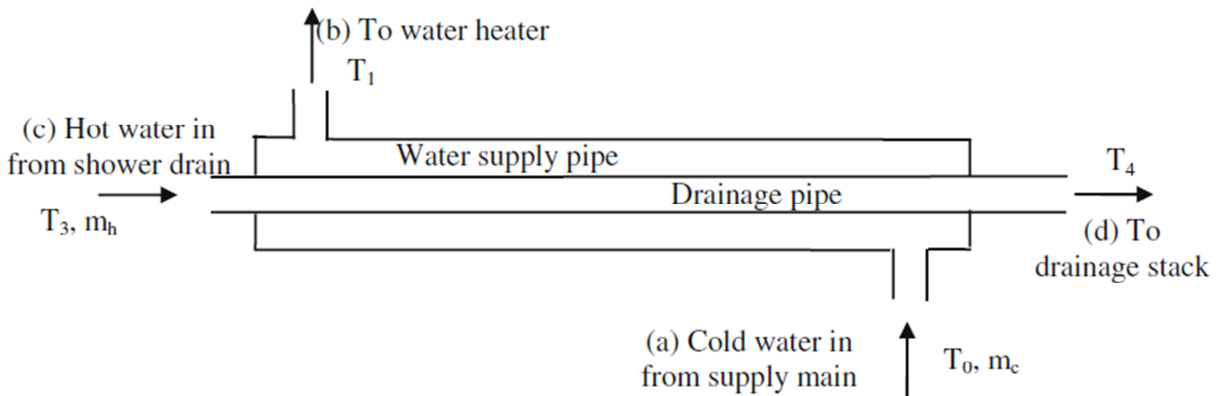


Figure 8. Schematic of heat exchanger used by Wong et al. (2010).

The energy savings result from the study by Wong et al. (2010) is calculated by using Monte-Carlo simulations. The input variables in the simulation include monthly average supply water temperature and the temperature drop from the shower head to floor drain. Also included, is a frequency distribution of the number of showers per person per day which was obtained

through survey results, along with another distribution for shower duration. Finally, the simulations were completed with a heat exchanger performance data set. This data set included a range from the worst case to best case overall heat transfer coefficient, as well as different pipe diameter and length configurations. A 1.5 meter heat exchanger with a 50 mm diameter yielded a 4-15% savings on water heating energy. Though not explicitly stated, it is assumed that the simulation input for the overall heat transfer coefficient of the heat exchanger is directly influenced by the performance data gathered for the heat exchanger tested in lab.

Another heat exchanger based heat recovery system was developed under a grant from the U.S. Department of Energy (DOE) Inventions Program. The device is known as a gravity-film heat exchanger (GFX). The GFX system is a counter-flow heat exchanger with a 3-4" center copper pipe and a 1/2" copper pipe coil structure wrapping around the larger pipe. The coil is flattened at the surface, where the two pipes mate to increase contact area. The GFX is designed to be installed vertically, thereby, allowing waste water to enter the large pipe diameter from above, while the heat sink fluid enters the copper coil from the bottom. A diagram of the GFX device is pictured in figure 9. The concept behind the GFX design is that as water falls through a pipe by gravity, it travels as a film over the inside diameter of the drain pipe while solids are able to pass unobstructed through the middle. This film of water is then in direct contact with the wall separating it from heat sink fluid (ORNL, 2005).

The GFX system is a non-regenerative device without a thermal storage system, and could potentially be installed in a distributed or centralized scheme. Despite being a passive system, the GFX requires a significant amount of vertical distance between the waste discharge source from which it recovers energy and the final discharge outlet of the residence. Figure 10 shows a diagram of a GFX installed as a centralized system in a home. The vertical space

requirement for the GFX may prevent it from being used in single story buildings without a basement, yet it remains a compact unit that can be installed in a wall cavity if the building structure favors the necessary conditions (ORNL, 2005).

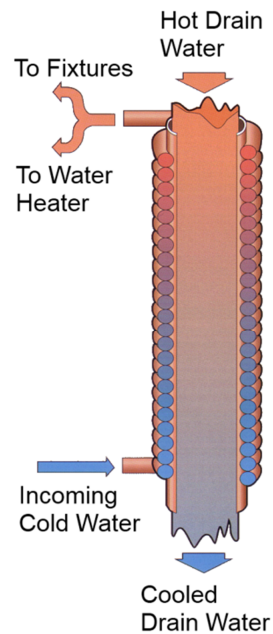


Figure 9. GFX diagram (CSG, 2004).

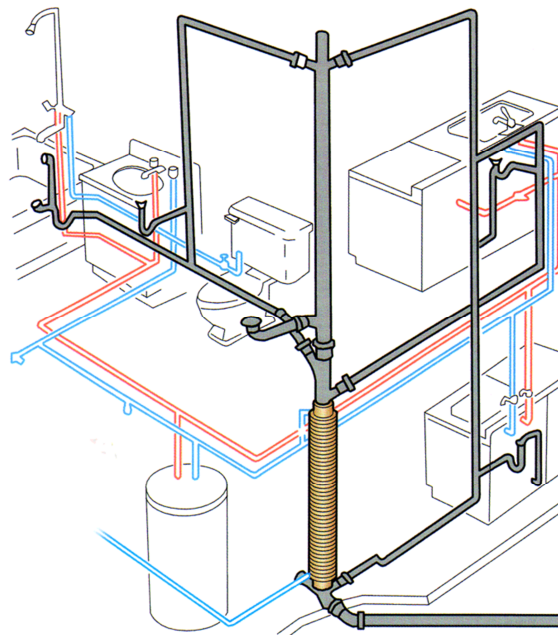


Figure 10. GFX installation (CSG, 2004).

A field test was completed by ORNL (2005) to test the performance of the GFX system. A 60" (assumed as the vertical height) version of the heat exchanger was installed in the basement of a single family home as a centralized system used to preheat the main water feed of the residence with the drainage from the entire home. Operating the shower at multiple temperature settings from maximum temperature (with no cold water tempering) down to 90 °F (32.2 °C), which was deemed the coolest reasonable shower temp, the study found that the GFX was capable of saving about 40% of the water heating energy needed for the shower over the range of temperatures. The study also conducted tests on possible installation configurations and the effects on preheating water with the recovered thermal energy from residential discharge water. Based off the experimental data collected (shown in figure 11), a model was created to simulate the effects of different installation configurations on the heat recovery performance of the GFX. Three different installations were investigated: balanced flow, unbalanced flow for preheating the cold water supply to the residence excluding the water heating system, and unbalanced flow for preheating only the water heating system cold water supply.

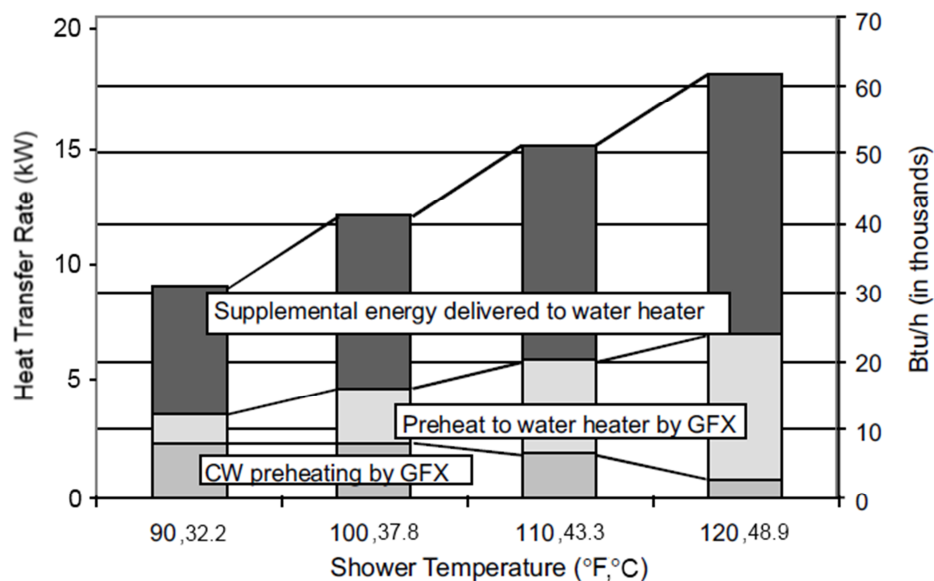


Figure 11. GFX performance data (ORNL, 2005).

Balanced flow is the condition in which the flow rates of both streams passing through the GFX are equal. In the ORNL (2005) study, balanced flow was the installation configuration for the experiment with the main water supply for the house routed through the GFX, as well as all the waste water drainage of the residence. This configuration, therefore, has the ability to preheat both the feed water going to the water heating system, as well as the cold water supply for the entire residence. Figure 11 shows the performance of the GFX for 60 °F (15.6 °C) influent inlet water temperature, 65 °F (18.3 °C) ambient air temperature, and a 2 gallons per minute (0.13 L/s) shower flow rate. As the shower temperature increases, the function of the GFX to preheat the cold water supply of the house becomes less dominant than the function of preheating the water to the water heating system. This transition is likely due to the fact that as the shower temperature increases, the flow from the water heating system increases, and therefore, more recovered energy is utilized in the water heating branch. Consequently, the reduced flow in the cold water branch carries less recovered energy. The balanced flow configuration demonstrates a negligible dependence on shower temperature to the overall energy recovery efficiency. However, it should be noted: if either plumbing route to the shower has greater heat loss potential, the overall recovery efficiency is likely to diminish in the regime where that route carries a higher initial percentage of the recovered heat. Also, if cold water draws are made from fixtures, aside from those that require tempering, there is a possibility to lose recovered heat without making any energy savings. However, in the absence of these parasitic cold water draws, the study simulations found the balanced flow condition to have the best performance of the other two configurations.

In the first unbalanced flow condition, the GFX can preheat the cold water supply to all fixtures in the house except the hot water heater. In the second unbalanced flow condition, the

GFX can preheat water only to the hot water heating system. Therefore, each unbalanced flow condition causes the drainage flow rate to exceed that of the cold feed flow into the GFX. This is due to inlet supply water that bypasses the GFX in both cases while retaining the drainage flow from each. The ORNL (2005) states that where a balanced flow installation is less practical, an unbalanced configuration that preheats the hot water system supply is favorable due to the absence of the chance for parasitic losses to cold water fixtures in a home. The simulation results also support the experimental data by showing that unbalanced flow, which preheats the water heating system stream, increases in water heating energy savings with increased shower temperatures, while the unbalanced flow for preheating the cold water supply decreases in water heating energy savings. Again, this is likely due to the change of flow rate through each branch as the shower temperature increases.

2.3.4 Heat Pump System

Another heat recovery system that has been published in literature is a study involving the use of a heat pump as a method to recover heat from waste discharge water. A study by Liu et al. (2010) investigated the use of a heat pump to recover waste heat from a public shower facility. The system investigated in this study is a centralized, non-regenerative system that also utilizes a heat exchanger to preheat the incoming supply water with the discharge water before the supply water continues to be preheated by the condenser of a heat pump which draws heat from the discharge water that leaves the heat exchanger. The system collects all discharge water from the shower drains into a grey water pool, from which the water is pumped through a filter before passing through the heat exchanger and evaporator en route to the final building discharge. After entering the building, the supply water is heated by the heat exchanger and then the condenser of the heat pump cycle, before supplying the water heating system. Despite being

a non-regenerative system by definition, given the described operation scheme, the use of a grey water pool gives this system an ability to store thermal energy and could, in fact, be operated as a regenerative system.

A heat pump is a vapor compression cycle that compresses a working fluid as a vapor to an elevated temperature and pressure which condenses in a condenser, thereby, releasing heat before being expanded to a lower temperature and pressure. At the reduced temperature and pressure, the working fluid in a saturated liquid-vapor phase continues to evaporate in the evaporator, thereby, absorbing heat while transitioning to a vapor for compression again. The advantage of a heat pump is the ability to heat and cool the fluids in contact with the evaporator and condenser with more energy than the mechanical energy necessary to run the vapor compression cycle. A heat pump is necessary to move heat from a colder fluid to a warmer fluid. This is possible when the condenser temperature is greater than the thermal sink, and when the evaporator is colder than the thermal source. A heat pump is more effective at moving heat from a low temperature to high temperature as the temperature gradient decreases.

In cases where electricity is favored for water heating, the use of a heat pump is advantageous over electric resistance heating because a heat pump has the ability to move more thermal energy across a temperature difference than the mechanical energy needed. Electrical resistance heating can be at best 100% efficient, therefore using more electricity than a heat pump operating with a coefficient of performance (COP) greater than unity. The COP is the ratio of thermal energy released by the condenser of a heat pump to the electrical or mechanical power required to enable the heat pump cycle. The advantage, which the study by Liu et al. (2010) is attempting to demonstrate by using a heat pump for waste water heat recovery, is that using waste water as the evaporator heat source can reduce the temperature gradient with the

condenser. This is a favorable idea in theory, assuming that the outdoor air temperature is cooler than the waste water; otherwise, the air would be a better source of heat since a heat pump is more effective when there is a lower temperature gradient across the evaporator and compressor.

The system studied by Liu et al. (2010) also included a solar water heating system as well as the heat recovery measures to reduce energy consumption demands necessary to heat water for the public shower facility. However, based on the predictions made in this study, the heating system with the heat recovery provisions aforementioned would require 44.5% less electricity than the solar and electric resistance water heating system does without heat recovery over 15 years of service. This estimate takes into account the electricity required to run the heat pump compressor and water pump needed to pressurize the grey water from the pool through the filter, heat exchanger, and evaporator. In fact, the electric resistance contribution is expected to decrease 96.2%.

2.3.5 Summary

Smith (1975) found that the greater the discharge flow rate compared to that of the inlet flow rate, the more effective the system is at raising the inlet temperature. Behrmann et al. (1982) also has results that support this behavior for the heat pipe system. However, Barbo (1985) reports that as the hot discharge flow rate increases above the inlet flow rate, the heat sink, being the inlet stream, becomes smaller than the heat source (the waste water stream). This allows significant temperature increases in the inlet flow, but reduces the heat recovery effectiveness. Results of simulations by ORNL (2005) support this condition as well. As the discharge flow exceeds the supply to the heat exchanger, less water heating energy is saved. Equation 1 is used to define the heat recovery efficiency for a two stream heat exchange. Equation 2 is the energy balance for a heat exchanger and assumes adiabatic conditions which

mean that no heat is lost or gained to the environment. Equation 2 shows that for a fluid with a larger heat capacity, it will have a lesser change in temperature in relation to the other flow. This means that a hot flow with an increasing heat capacity will reduce the numerator due to a decreasing temperature difference if the percent decrease of the hot flow temperature differential is greater than or equal to the percent increase in the hot flow rate. This will cause the heat recovery efficiency to decrease.

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{(\dot{m}c_p)_H(T_{H,\text{in}} - T_{H,\text{out}})}{(\dot{m}c_p)_{\min}(T_{H,\text{in}} - T_{C,\text{in}})} = \frac{(\dot{m}c_p)_C(T_{C,\text{out}} - T_{C,\text{in}})}{(\dot{m}c_p)_{\min}(T_{H,\text{in}} - T_{C,\text{in}})} \quad (1)$$

$$[\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_C = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_H \quad (2)$$

Behrmann et al. (1982) also discovered that under balanced flow conditions with similar waste inlet temperatures, energy recovery efficiency decreases as the flow rate increases. The authors suspected that a limit in heat flux is attained at high flow rates. In fact, the heat transfer by heat pipe may be subject to four major limitations, including a sonic, entrainment, wicking, and boiling limit (Mills, 1995). The effect of these limitations on the performance of a heat pipe can be seen in figure 12.

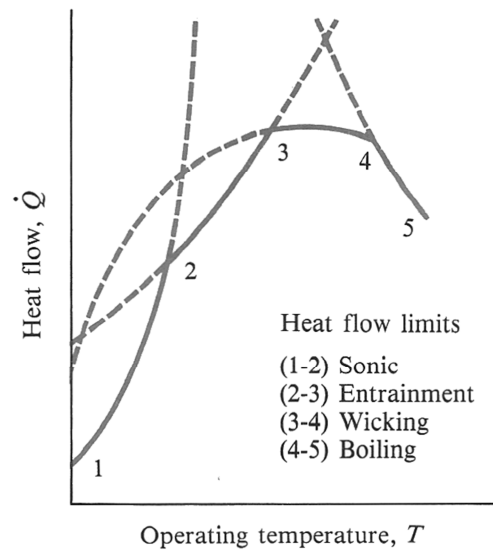


Figure 12. Heat flow limits for a heat pipe (Mills, 1995)

Given the apparent heat flow limitations of a heat pipe, there is a required need for caution when designing a system with this type of element. These potential confinements need to be taken into consideration.

Given the many systems that have been developed in different studies, it is apparent that there are a number of heat recovery options to consider for utilizing residential waste water. From the results of these systems, it is shown that an energy savings of anywhere from 4-44.5% can be obtained for water heating depending on the system utilized. However, there are few endeavors in the study of residential heat recovery despite the promising results. This study will help develop the research field so as to uphold the energy conservation responsibility.

CHAPTER 3: METHODOLOGY

3.1 RESEARCH PURPOSE

The goal of this study is to further help characterize the potential value in residential waste water heat recovery systems. This study will explore the potential heat recovery of a non-regenerative, centralized waste water heat recovery system that is used to preheat the main water supply of a residence. Using the results from experimentation, the accuracy of prediction methods will be analyzed and evaluated for limitations. This study will help to identify and reaffirm the factors that influence the potential for residential waste water heat recovery.

3.2 EXPERIMENT DESIGN

This study investigates the potential for waste water heat recovery in a house that was designed and built for the 2009 Solar Decathlon competition. The house built for the competition was required to be powered entirely by solar energy with intentions of being net-zero which means that it generates at least as much electrical energy as it consumes. Therefore, this grid-tied house utilizes a photovoltaic array to generate electrical power for use in compliment to the grid power. There is no solar thermal water heating generation included in the design of the house. Another key feature of the house design is the compliance with passive house design standards which allow the home to take advantage of solar radiant heating to minimize active system heating loads. Given the characteristics of this test home, an appropriate waste water heat recovery system was designed to help further reduce the energy consumption of the house.

It was decided by qualitative analysis that the best use for the recovered thermal energy would be to preheat the inlet supply water. This is a favorable practice from the standpoint that

domestic hot water is commonly utilized by inhabitants of a typical home throughout the year. This contrasts the other prominent concept to use the recovered thermal energy to aid the space heating function of a residence which is primarily beneficial during the heating season only. This is still not an obvious choice because if the system used to heat water and the system used for space heating have different levels of performance enhancements by using the recovered heat, this would cause less dependence on the usage frequency. If the heating load of a home is large enough and the performance increase from utilizing the recovered heat is great enough, then it could cause a greater reduction in annual energy consumption despite being in operation for a less amount of time in a year compared to the hot water system. However, considering the grade of thermal energy to be recovered from the waste discharge, the space heating system would only be able to benefit if the temperature was high enough for direct heat exchange, otherwise it would have to supply heat to an evaporator in a heat pump. The heating system on the 2009 Solar Decathlon house is a heat pump system and could significantly benefit from the waste heat used at the evaporator during the winter months when the temperature gradient is at the highest point, but this system would require significant parts and installation cost as well as the development of more sophisticated controls in order to integrate it with the existing system. To address the temporal disconnect between waste heat availability and space heating needs, a thermal storage system would be required. Though utilizing the waste water heat for space heating could prove more beneficial to overall energy savings, it has a significant cost associated to a retrofit situation. Also, due to the fact that the home is designed for passive house standards which already cause a reduction in active system heating load, it is questionable if the performance enhancement to the space heating system would overcome the savings of a water preheating system with a greater usage frequency.

Additionally, a non-regenerative and centralized heat recovery scheme was chosen for the experiment. The centralized scheme was chosen due to the fact that it was a retrofit system and would require less specialized installation requirements. A non-regenerative scheme was chosen based on data published by Lowenstein and Hiller (1998) and ORNL (2005) that attributes 41.6% and 43% of the hot water consumption in a residence specifically to shower demands. In each study (results found in table 1), shower consumption is the largest portion of the hot water use, and amounts to more than the other fixtures and appliances that benefit from regenerative systems (dishwasher, clothes washer, and baths) combined. The ‘other’ category in table 1 is mostly sink usage which also benefits from the non-regenerative system. Specifically targeting waste heat recovery from sources that do not require regenerative recovery schemes eliminates the need for thermal storage while still targeting a majority of the hot water for heat recovery. Thermal storage is avoided for the particular experiment site because it is only a 1000 sq. ft. home designed for two people which therefore increases the intermittency between hot water accumulation events such as dishwashing and clothes washing (there is no bath in the 2009 Solar Decathlon house) which would increase storage time heat losses and decrease the recovery effectiveness.

Table 1. Residential Hot Water End-Use Disaggregation.

End-Use	(Lowenstein and Hiller, 1998)	(ORNL, 2005)
Dishwasher	10.8%	13.0%
Showers	41.6%	43.0%
Clothes Washing	12.6%	20.0%
Baths	8.0%	6.0%
Other	27.0%	18.0%

A regenerative system may prove more beneficial if the demand intermittency is low enough that tank losses have less detriment than the added ability to recovery heat from all hot

water fixtures and appliances. It may also prove beneficial if the temperature and flow of the drainage from the showers and sinks is great enough that it is more cost effective to transfer heat at a lower rate than a faster non-regenerative method. The regenerative method does, however, likely require increased space requirements, large valves resistant to fouling and clogging, and a control system. It was still determined that a non-regenerative system is a better choice for the 2009 Solar Decathlon house as space is not available in the home for additional water storage. Also, the flow rates of the shower and sinks are suspected to be within reason for a non-regenerative system to be more cost-effective compared to a regenerative system.

A counter-flow flat plate heat exchanger was chosen as the method of heat recovery in the centralized, non-regenerative system. The choice to use a heat exchanger was made due to the common availability of several proven design types and sizes as well as for the depth of theory that exists for these devices. This made the heat exchanger a favorable choice over the thermosyphon and heat pipe but it was still picked over a heat pump system for its limited complexity and as Liu et al. (2010) describes that the heat exchanger has a lower cost per unit of recovered energy under temperature limitations when compared to a heat pump. A counter-flow heat exchanger was chosen because for a set NTU and R_c a counter-flow heat exchanger is more effective than a parallel flow heat exchanger due to a larger temperature differential that is maintained over the length of the exchanger (Mills, 1995). The flat plate heat exchanger was chosen under a special condition related to the application of this system on the 2009 Solar Decathlon house. For the competition, the toilet was capped and rendered non-functional. This was also true for the experimentation done in accordance with this study as the home was not being used as living quarters. This also means that no other solid particles should be put down the drainage system during the testing process. Given this condition, a flat plate heat exchanger

which would normally be susceptible to clogging in the event of any large solid discharge could be utilized with minimal filtering. A flat plate heat exchange was preferred for the study given the inherent high heat flux ratings and therefore limited required space and higher expected cost effectiveness.

With the results published by ORNL (2005) that for non-regenerative heat recovery systems, the balanced flow condition is the most effective at recovering energy, it was decided to utilize this configuration. The flat plate heat exchanger will be located in the stream of the waste discharge while the other stream is provided by the water supply line to the house. This allows recovered energy to preheat the cold water supply of the house as well as the supply water to the water heating system which is a heat pump water heater with a storage tank. This provision ensures that the waste stream flow rate does not exceed the inlet stream flow rate which would reduce the heat exchanger effectiveness as previously discussed.

3.2.1 Apparatus

The placement of a flat plate heat exchanger in the waste water discharge line presents flow impedance and therefore a requirement for a pumping method to maintain the discharge of water and prevent water backup inside the home. A system was developed similar to that utilized in the study described by Liu et al. (2010). Water that drains from the house under atmospheric pressure by gravity flow collects in a small catchment pool from which water can be pumped through a filter and into the heat exchanger. From the heat exchanger the water is channeled back into the sewer flow. All discharge is channeled through the heat exchanger when the flow is below the threshold of the pump capacity, and exceeding the threshold causes flow to overflow the catchment pool and bypass the heat exchanger.

The design of the 2009 Solar Decathlon house conveniently located the 4" PVC discharge water pipe from the house within a couple feet of the location where the 1" PEX supply water line enters the house. Because this is a one story house and the discharge pipe is slightly above grade, it was not possible to employ a GFX type heat exchanger without having to dig a deep hole and create a sump pump system to elevate the water back to grade after dropping through the heat exchanger. A note about the water utility at the experiment site is that all water is supplied by a jet pump with a pressure tank from a clean water reservoir while all discharge water is collected in an identical storage vessel each located above grade. With the discharge water collection tank (henceforth referred to as the grey water storage tank) located above grade, and above the height of the discharge line from the house, it required a sump pump system to collect the house discharge water and pump it into the grey water storage tank. This water system configuration was designed for use in the competition which was held at a site without access to public utilities (water and sewer). Despite already having a sump pump in the discharge system, a GFX was still avoided for the vertical limitation requiring a deep hole. It is also of value to test a different heat exchanger concept and provide unique data to the field of study related to residential waste water heat recovery.

Figure 13 and 14 depict a diagram of the apparatus used for experimentation. Water leaving the house is channeled into the main branch of a PVC y-pipe in the opposite direction of intended flow. The main branch of the y-pipe is capped at the other end causing the water to stagnate and pool. The capped end of the y-pipe is blocked by reducer bushings that open to a smaller inlet to a diaphragm pump which pressurizes the discharge flow to push it through the flat plate heat exchanger. After passing through the heat exchanger, the cooled waste discharge is piped back into the main discharge pipe line downstream at a lower elevation to prevent

recirculation. The supply water stream enters the heat exchanger at the same side that the discharge stream exits and flows in the opposite direction through the heat exchanger as the discharge stream until the warmed water exits and enters the house. Due to the fact that the discharge pipe and supply inlet for the house are located just above grade, the entire apparatus had to have a 2 ft. deep hole created with a perimeter of approximately 3 ft. x 2 ft. to fit in place properly. If discharge flow from the house exceeds the rated capacity of the pump, the water level in the y-pipe pool will raise and spill over into the part of the discharge pipe that leads to the grey water storage tank and bypass the heat exchanger. The heat exchanger was however sized to accommodate the flow from all showers and sink fixtures simultaneously.

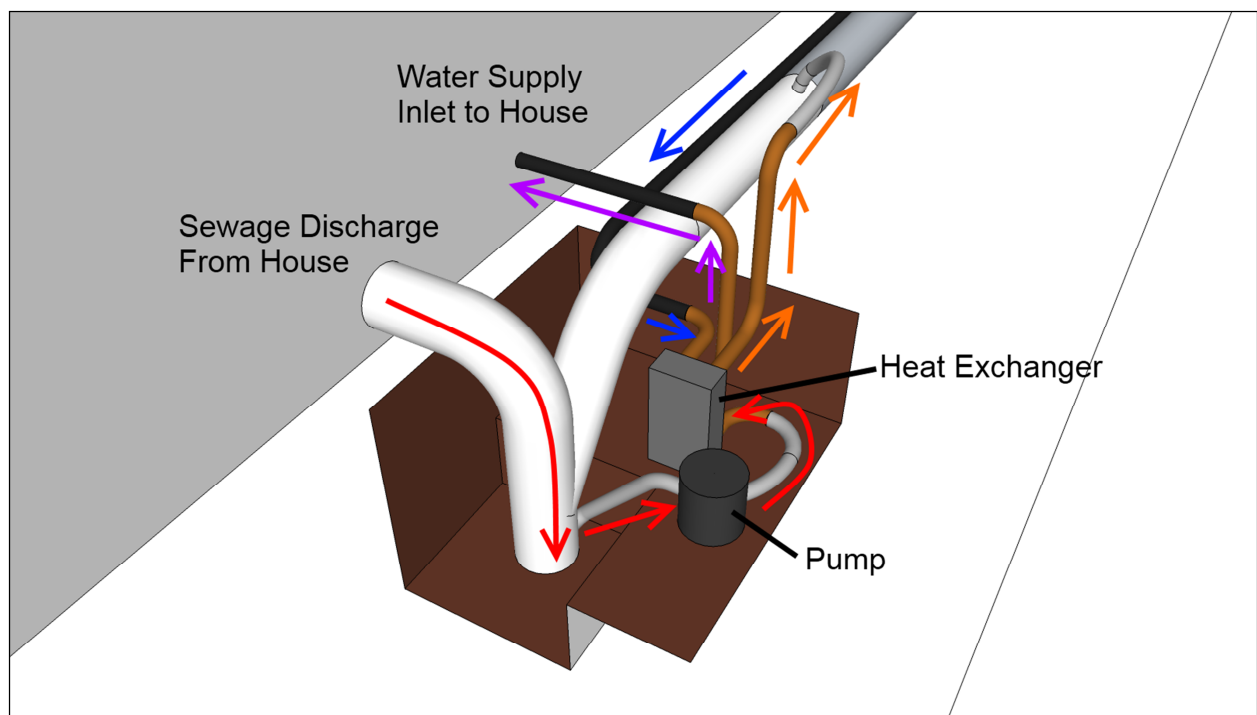


Figure 13. Left view of experiment apparatus diagram.

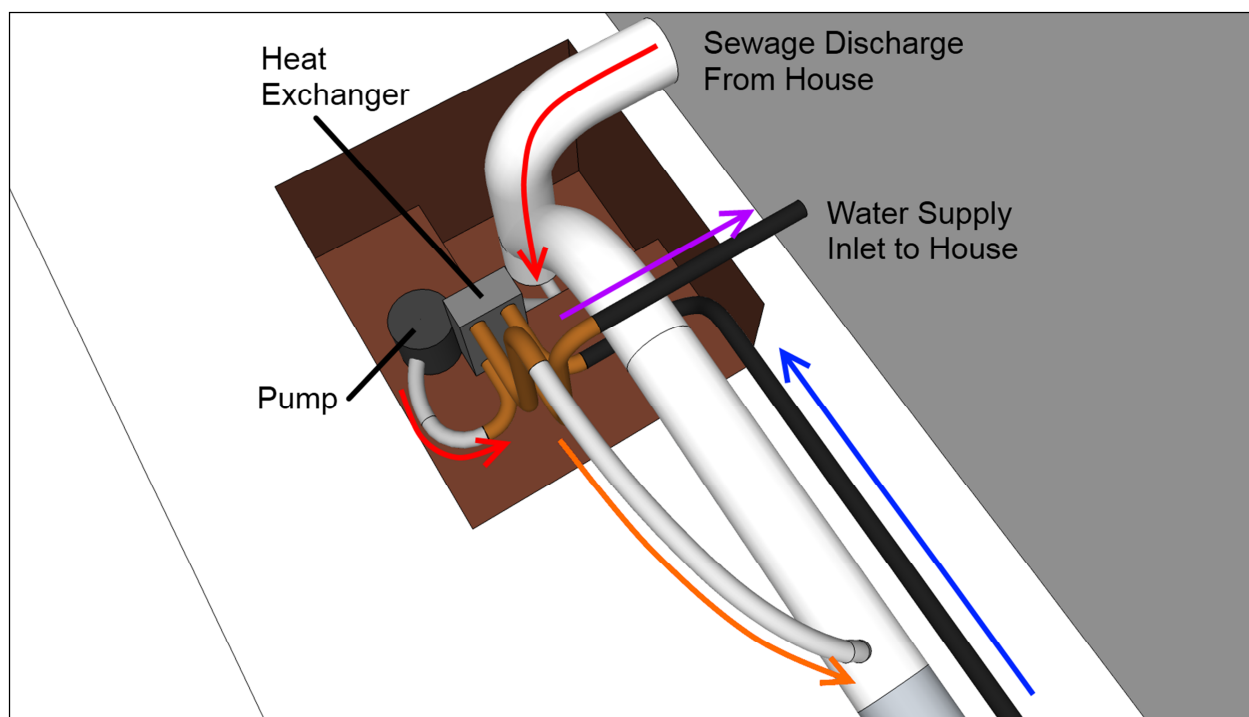


Figure 14. Right view of experiment apparatus diagram.

Figures 15 and 16 are photographs of the heat recovery system installed at the test site. T-type thermocouples (copper and constantan) are installed and located at the entrance and exit of the heat exchanger in both influent and effluent streams. A flow meter is installed on the outlet side of the heat exchanger for both streams. All lines that potentially carry water at an elevated temperature over ambient conditions are insulated to prevent heat loss with 3/8" thick polyethylene pipe insulation. A diaphragm pump was utilized in this experiment for its ability to self-prime and to be run dry, therefore limiting the amount of control necessary to protect the pump from damage. The location of the pump is closer to the house than shown in figures 13 and 14. To control the flow of the discharge water through the heat exchanger, two globe valves were used to throttle the flow rate. This is necessary to match the flow rate of the discharge flow to that of the water supply flow rate. Though one globe valve installed on the outlet side of the heat exchanger in the discharge stream should be sufficient to reduce the flow, it was found that

in order to match the rate of the supply water the valve throttled the flow from the pump too much causing the pressure switch cut-off to activate on the pump. To counteract this effect, a bypass line was installed on the inlet side of the heat exchanger in the waste water stream to allow mass flow from the pump to recirculate and further help control the mass flow through the heat exchanger.

Also included in the test apparatus were three T-type thermocouples that were placed in the house to measure the water temperature of the water coming out of the shower head and out of the kitchen sink as well as the water entering the shower drain. All temperature and flow measurements were recorded using National Instruments LabVIEW (Laboratory Virtual Instrumentation Engineering Workbench) software.



Figure 15. Side view of test apparatus.



Figure 16. Top view of test apparatus.

3.2.2 Test Procedure

Three separate tests were conducted, each monitoring the heat recovery performance of the flat plate heat exchanger for three separate flow rates. The first test monitors performance with only the shower running at the highest temperature and flow setting. Next, a test was conducted with the addition of the kitchen sink running at maximum temperature and flow. Then the final test was conducted with the addition of the shower wand accessory functioning simultaneous to the shower and kitchen sink, each on the highest temperature and flow setting.

Each test began by starting cold water at the appropriate fixture with maximum flow rate and starting the diaphragm pump. The throttling valves on the discharge stream were then set to match the flow rate of the incoming supply rate. Precaution was taken to ensure that all air was

purged from the pump feed by circulating water through the bypass tube while a small pool of water formed in the y-pipe. This was necessary to ensure an accurate flow reading of water flow and not water flow with entrained air bubbles. After purging the discharge line of air and setting the flow rate of the discharge equal to that of the supply, the appropriate fixtures were switched to the hottest water settings. While the test ran, constant monitoring of the flow rates was necessary to prevent the pool in the y-pipe from depleting or overflowing due to an unbalance flow condition.

CHAPTER 4: RESULTS AND DISCUSSION

4.1 TRANSIENT BEHAVIOR

The first test was conducted by running only the shower in the test home with an initial indoor temperature of approximately 80 °F (26.7 °C). The HVAC system of the home was not operational throughout the duration of all three tests and therefore the indoor temperatures recorded are higher than what is expected for a normal home. This fact is not expected to adversely affect the heat recovery performance and is merely stated to help understand and interpret the data collected.

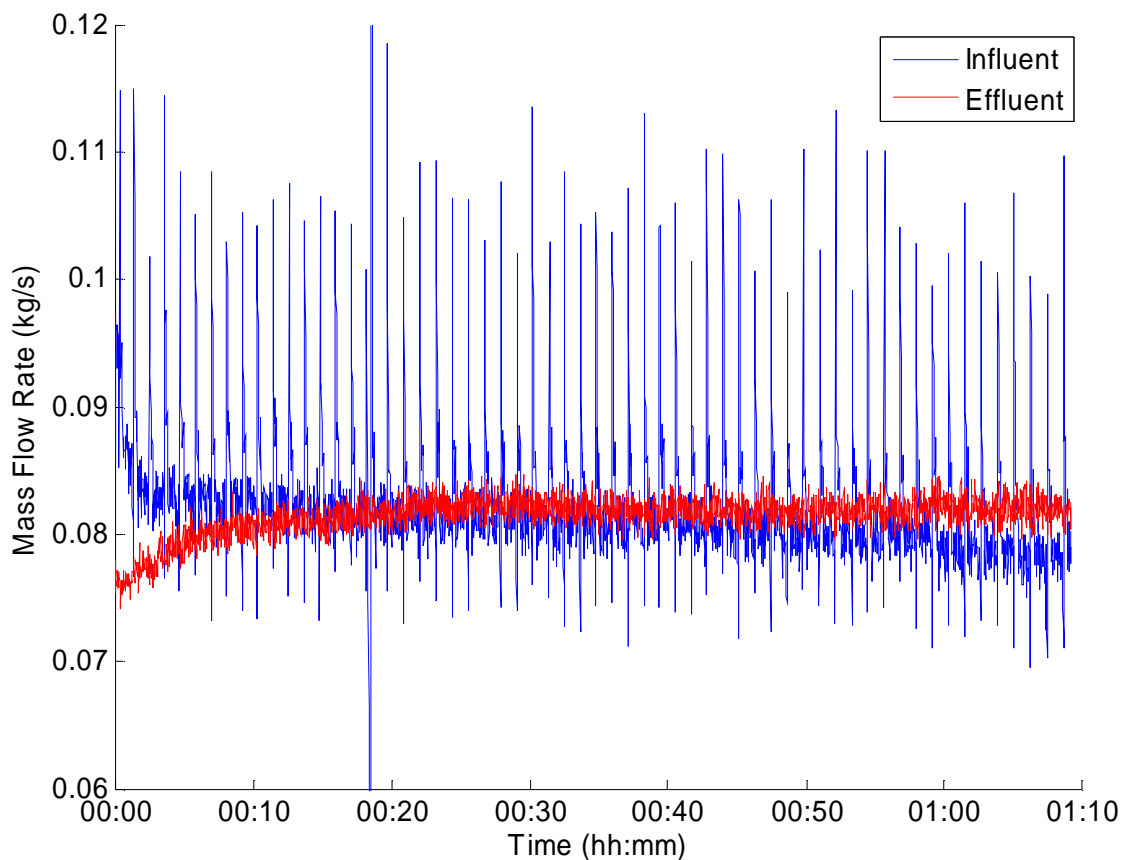


Figure 17. Test 1 water flows versus time.

Figure 17 reveals the flow rates of the supply water and discharge water for the duration of the first test. It is shown that the inflow rate from the supply water tank required to operate the shower is approximately 0.08 kg/s. The actual average flow rates for each test are tabulated in table 2. The supply flow rate indicated in figure 17, as well as all the test flow versus time flow plots, shows a cyclic spiking behavior. This is caused by the jet pump that is used to create the water pressure for the plumbing system for the test house. The jet pump has a pressure storage tank that is pressurized when it drops below a low threshold by initiating the pump which shuts off when the tank reaches the upper pressure threshold. When the jet pump initiates, the increased pressure causes the inlet flow rate to increase temporarily for the duration of the pressurization cycle. More detail about this phenomenon can be found in Appendix A. Another explanation about this flow data, related to the drastic drop and spike at approximately 18 minutes into the test, is that the pressure pump power was temporarily cut-off accidentally. The line pressure was therefore allowed to drop below the low pressure threshold of the pressure pump, therefore dropping the inlet flow rate well below the previous average. This mistake was caught within seconds of the occurrence and only accounted for a small fraction of the data set and therefore was deemed acceptable to continue the test. It is unclear what causes the supply flow to decrease over the duration of the test and for the discharge flow to increase in the same time. This phenomenon was noted and taken into consideration when setting the throttling for the outlet flow rate which is why the discharge flow rate is not equal to the inlet flow initially. This was also done to avoid un-priming the discharge hoses between the pump and heat exchanger and introducing air into the system which would cause unwanted variations in the heat transfer properties of the heat exchanger. The risk of doing this is that the pool in the y-pipe could fill and water would overflow and be lost for heat recovery purposes, therefore creating an

unbalance flow condition. The condition of cold flow specific heat exceeding the hot flow specific heat would, however, lead to increased heat exchanger effectiveness.

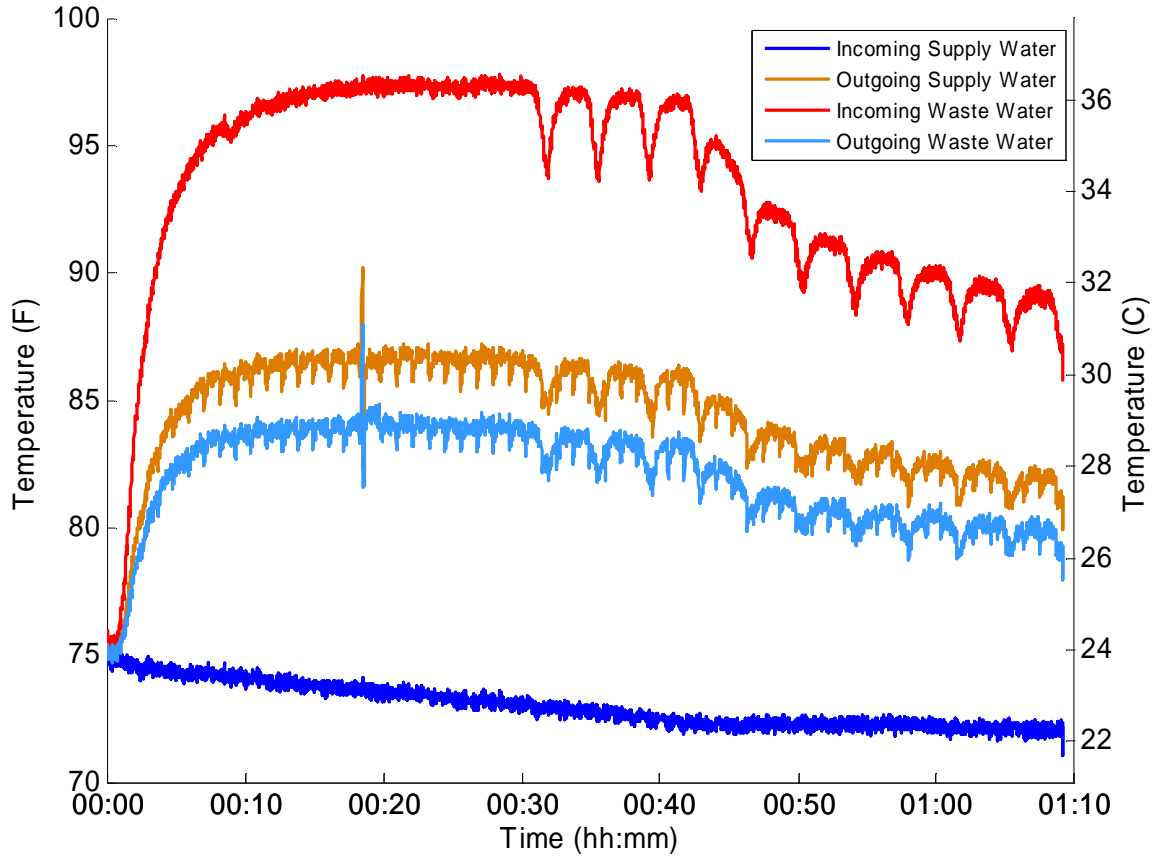


Figure 18. Test 1 temperature variations with time.

Figure 18 shows the temperature variation in the inlet and outlets to the heat exchanger. It can be seen that, initially, the waste water supply to the heat exchanger is at an increase temperature to the incoming water supply and the other two flow temperatures. This is likely caused by evaporative cooling of the inside air when the shower was running on the cold setting during the outlet flow throttling process. The incoming supply water temperature is another aspect to note about this plot and the similar plots to follow. Because this house has an above ground water supply tank, the supply water temperature is affected more by the outdoor air temperature in comparison to water from a buried supply line. This fact can be beneficial to the

water heating system in the summer (which is when the tests were conducted) because there is less of a temperature gradient required to charge the hot water tank. Conversely, in the winter, this is a detriment as the water would be cooler than the ground temperature and therefore require heating over a larger temperature gradient to charge the water tank. It is also obvious that without insulated outdoor pipes and water tanks, this system cannot operate in sub-freezing temperatures of the winter.

Asymptotic heating (where temperature increases to a steady-state value) is observed in all flows except the inlet from the supply tank as expected. The counter-flow nature of heat exchanger is apparent with the outgoing cold water supply temperature exceeding the outgoing waste water temperature. The incoming waste water temperature is observed to approach and steady at approximately 97 °F (36.1 °C). From figure 20 it can be seen that the shower drain temperature is approximately 100 °F (37.8 °C) at steady state conditions therefore indicating an approximate 3 degree temperature loss in the plumbing between the drain and the inlet to the heat exchanger. There is an odd phenomenon in figure 18 that is revealed just after 30 minutes with the onset of a cyclic behavior of cooling temperatures from the waste water flow inlet followed by a reheating with an overall time of approximately 2 to 3 minutes. This behavior remains undiagnosed but does not manifest as severely in subsequent tests.

The amount of heat recovered from the outgoing waste water discharge is plotted in figure 19. The heat recovered is the heat capacity of the cold flow multiplied by the temperature gradient across the supply water inlet and outlet of the heat exchanger. A derivation of the energy balance for a heat exchanger can be found in Appendix B. Figure 19 also reveals the heat loss over the duration of the test. It is apparent that the heat loss for this test does regularly exceed 250 W, or less than 10% of the steady state recovered heat rate of approximately 2500 W.

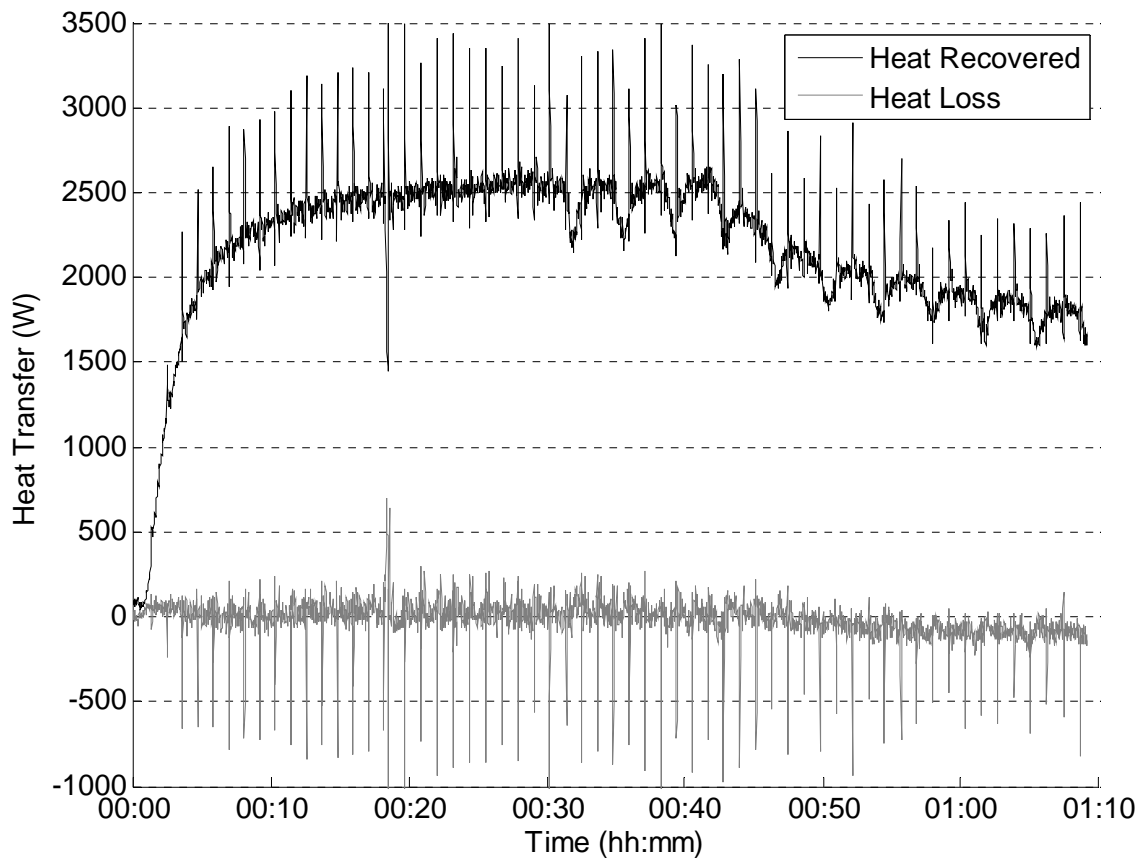


Figure 19. Test 1 recovered heat versus time.

Figure 20 is a plot of the indoor air conditions including temperature and humidity. Also included in the plot are the water tank temperature and the shower temperature at the spout and at the drain. The temperature stratification in the water tank is apparent by the decrease in water tank temperature by approximately 20 °F (11.1 °C) starting at 30 minutes into the test. The temperature does not drop at the shower head for about another 10 minutes. This is likely due to the fact that the water tank thermocouple is approximately 18-24” inches from the top of the water tank where the hot water outlet is to feed the shower. This figure also reveals that the water temperature drop in the shower is approximately 12 °F (6.7 °C) and that the relative humidity in the house rose about 15% to about 70% in the duration of the test.

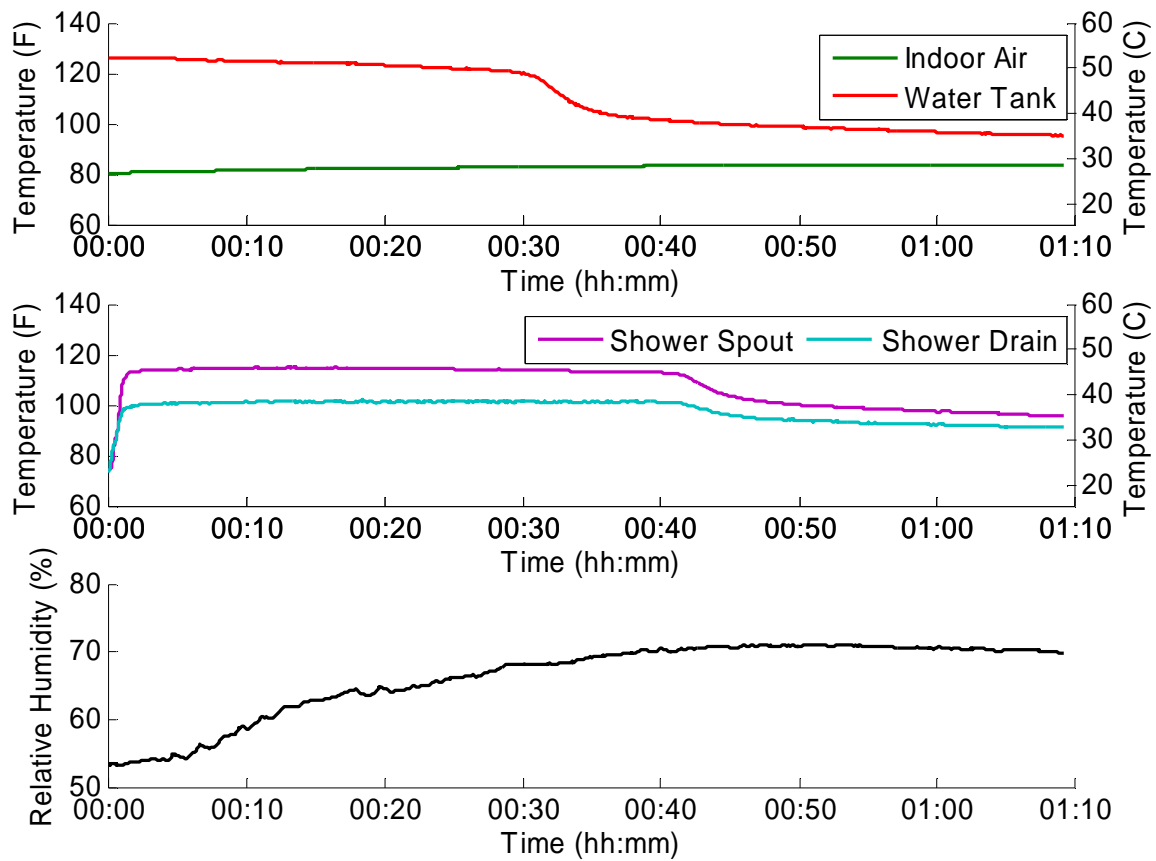


Figure 20. Test 1 house conditions, shower temperature, and water storage temperature.

The flow behavior of the second test was similar to that of the first test with an average inflow and outflow rate of approximately 0.11 kg/s as pictures in figure 21. The difference in the second test is the addition of running the sink concurrent to the shower to increase the flow rate. As expected, the increased flow rate shortened the duration of the test as the hot water supply depleted faster than for the first test case.

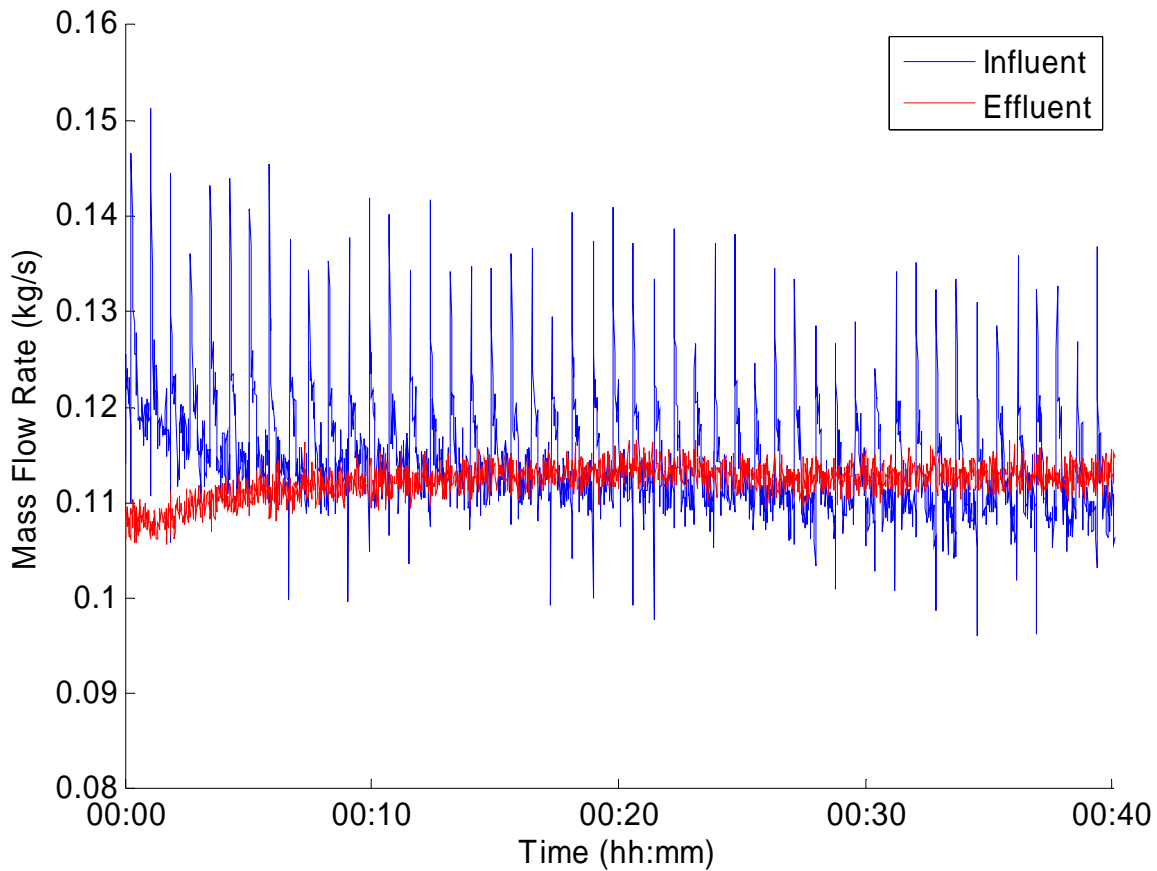


Figure 21. Test 2 water flows versus time.

The temperature variations with respect to time presented in figure 22 for the second test are similar in shape to those of the first test as expected. The incoming water temperature for this test is higher than the first at approximately 76 °F (24.4 °C). The variation in start temperatures is greater for this test due to the increased indoor temperature at the start of the test at approximately 90 °F (32.2 °C). This, coupled with the time required to throttle the outflow to match the inflow rate while running cold water through the fixtures, allowed time for the waste water temp to increase from evaporative cooling effects. The asymptote of the outgoing supply water is nearly 100 °F (37.8 °C). This is approximately the same steady-state temperature of the shower drain temperature indicating an even lesser degree of heat loss in the drainage pipe than the first test.

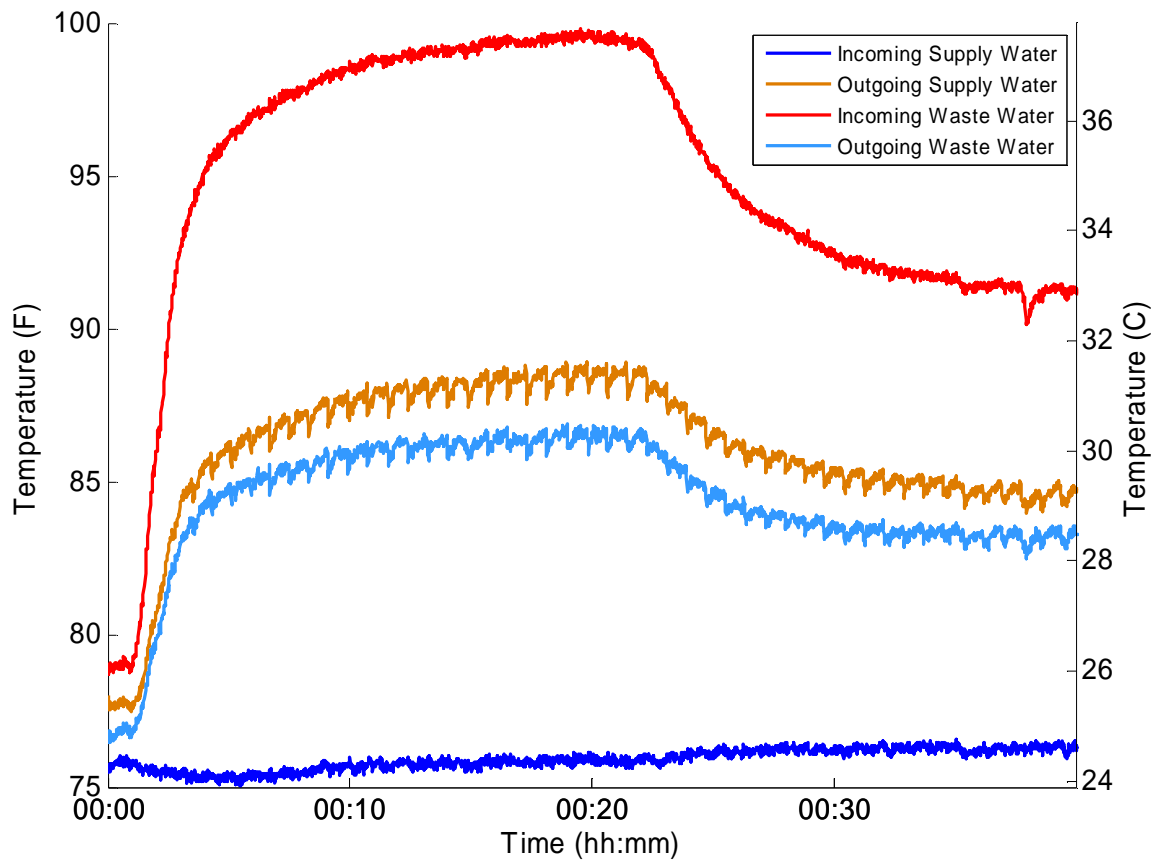


Figure 22. Test 2 temperature variations with time.

The recovered heat in the second test was observed to occur at a greater rate than in the first as is expected given the increased flow rate. The result pictured in figure 23 shows that the recovered heat transfer approaches an asymptote of approximately 3300 W again with heat loss to the environment remaining below 250 W or less than 7.5% in this case.

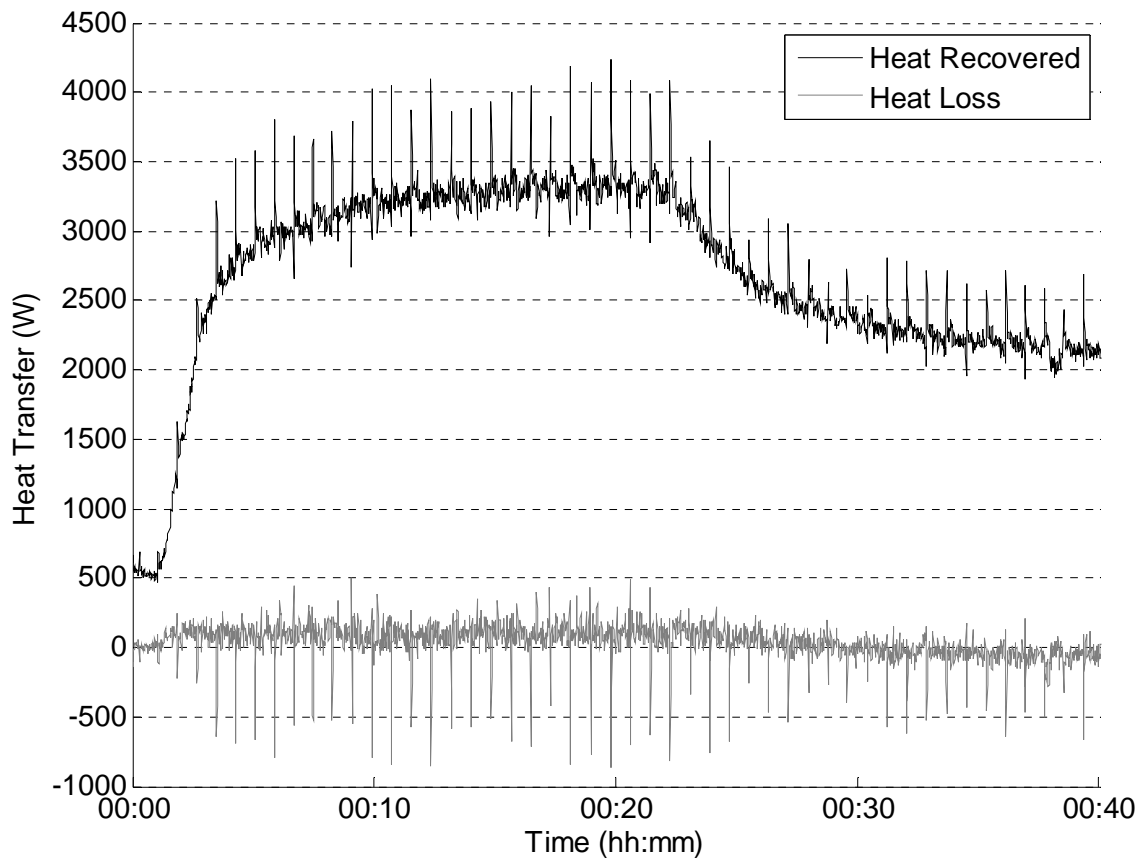


Figure 23. Test 2 recovered heat versus time.

The results pictures in figure 24 reveal the slightly more adverse test conditions endured for the second experiment with an indoor temperature of approximately 90 °F (32.2 °C) and relative humidity starting at 60% though only increasing to the mid 60's. The hot water tank only endured desired spout temperature and sink temperature at the additive flow for approximately 21 minutes. The initial hot water tank temperature for this test was better than 10 degrees lower than the first test which further inhibited the lasting duration of the hot water supply. The temperature drop from the shower spout to the drain for the second test was about 10 °F (5.6 °C) which is slightly less than that of the first test.

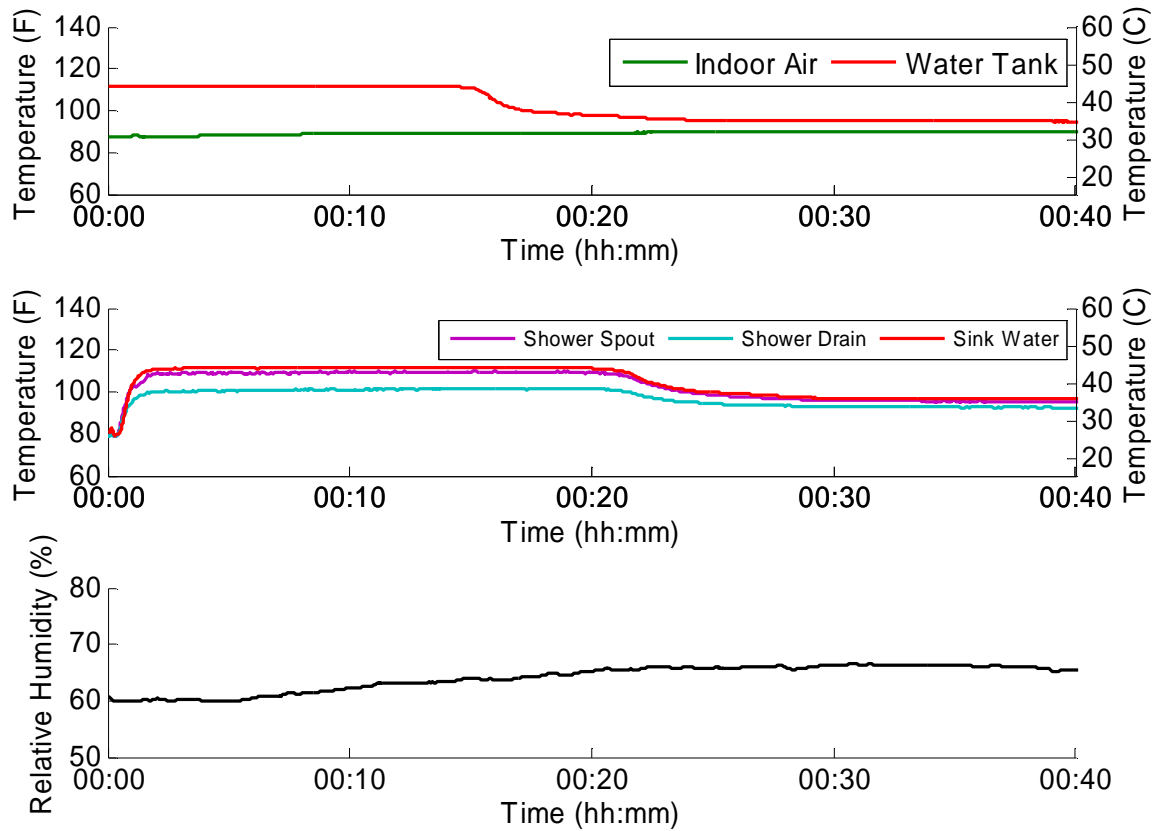


Figure 24. Test 2 house conditions, shower temperature, and water storage temperature.

The final test was completed with the shower running concurrent to the sink and a hand wand sprayer in the shower. The wand utilized the same tempering valve as the shower and is therefore assumed to be at the same temperature as the shower spout. Figure 25 shows the results of the flow variation with time for the duration of the third test. The average flow rates for the third test were approximately 0.19 kg/s. During the middle of this test, just before 20 minutes in, the inflow underwent an abrupt and significant increase to a higher flow rate relative to the preceding average flow. This variation is unexplained.

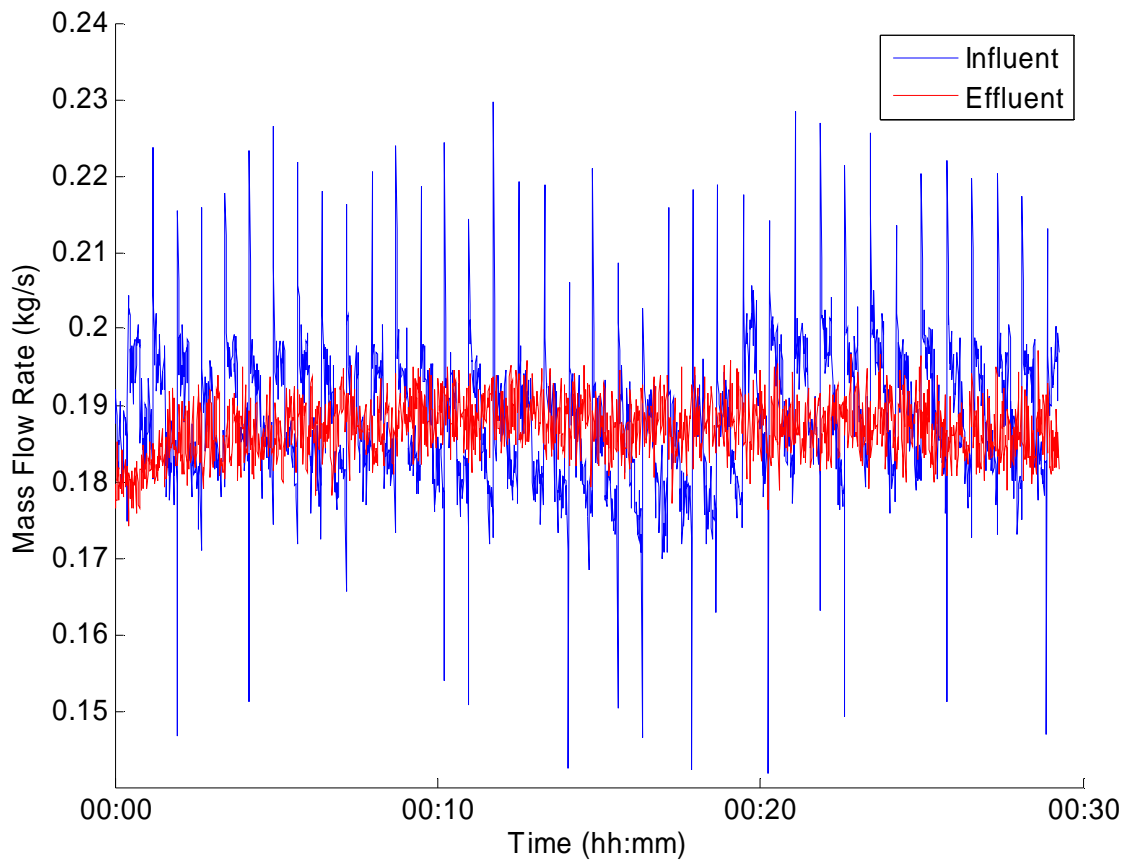


Figure 25. Test 3 water flows versus time.

The results of the temperature progression with time for the third test are similar to those of the previous two tests. From figure 26 it can be seen that the incoming waste water temperature asymptotically approaches 100 °F (37.8 °C) which is approximately the same temperature of the shower drain temperature. This again indicates minimal heat loss in the house plumbing at steady-state conditions.

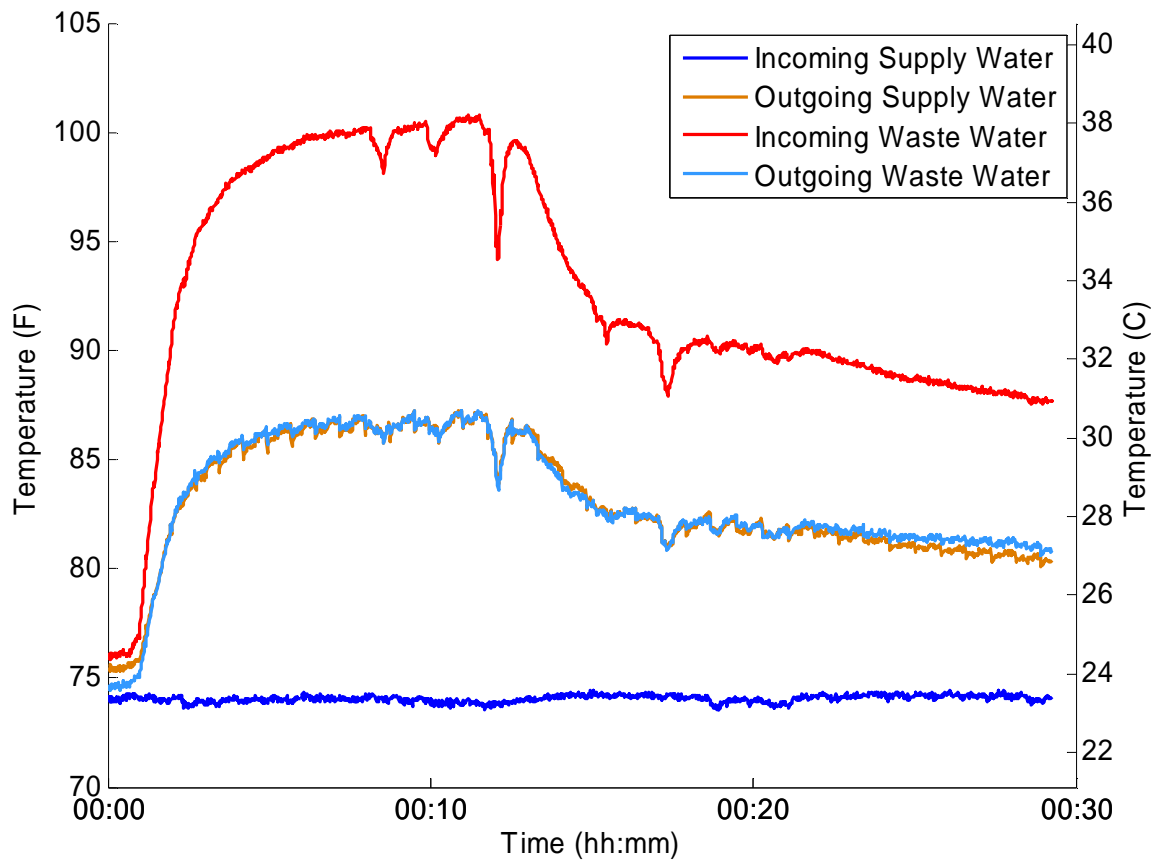


Figure 26. Test 3 temperature variations with time.

The rate of heat recovery in the third test is shown in figure 27. It was observed to be greater than that of the previous two tests and asymptotically approaches 5500W. The heat loss, however, shows a tendency to regularly approach 500W on the upper end which is almost 9% of the recovered heat rate.

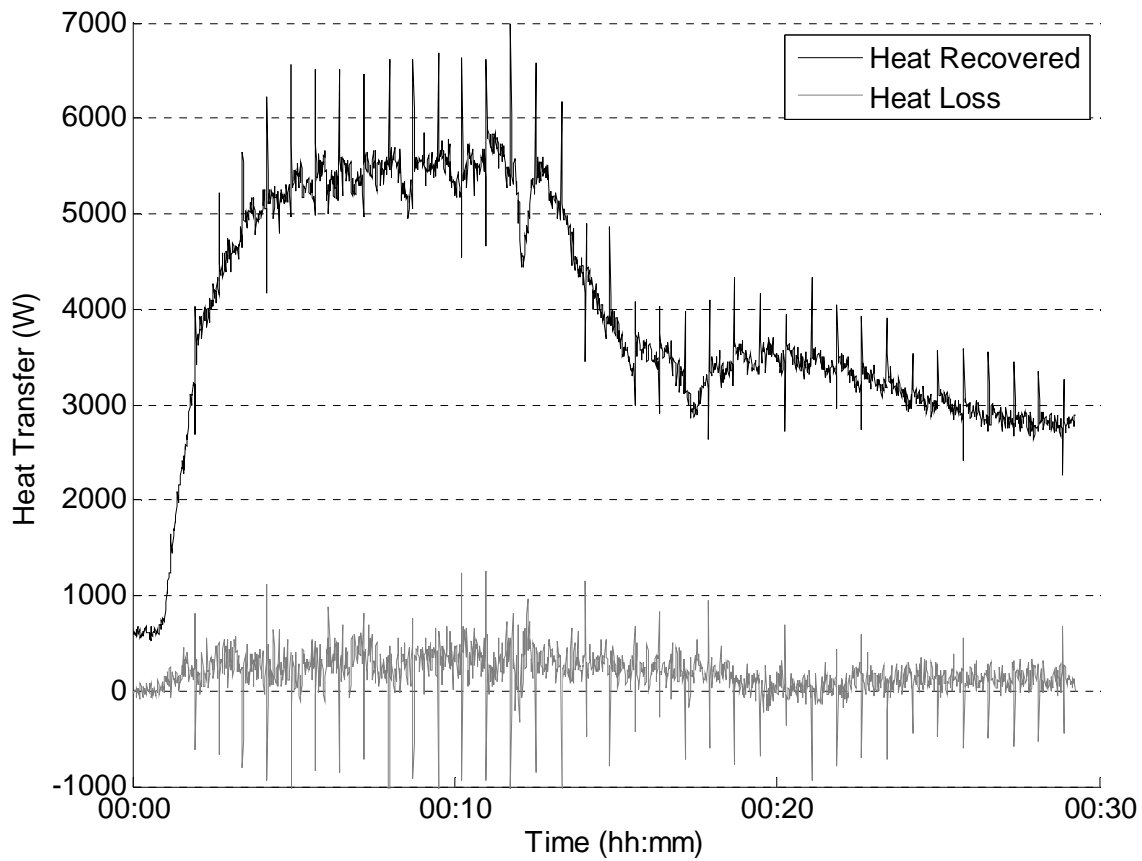


Figure 27. Test 3 recovered heat versus time.

The indoor environmental conditions of the third test proved similarly adverse to those of the second test. From figure 28 it can be seen that the initial indoor air temperature was in the upper 80's with a relative humidity in the mid 50's but rose to the mid 60's as the temperature moved into the 90's. The initial water tank temperature in this test was only slightly higher than that of the second test at approximately 115 °F (46.1 °C). The temperature drop of the shower temperature in the third test is approximately 10 °F (5.6 °C) at steady-state conditions, similar to that of the second test. The hot water supply at the increased flow rate lasted for approximately 12 minutes.

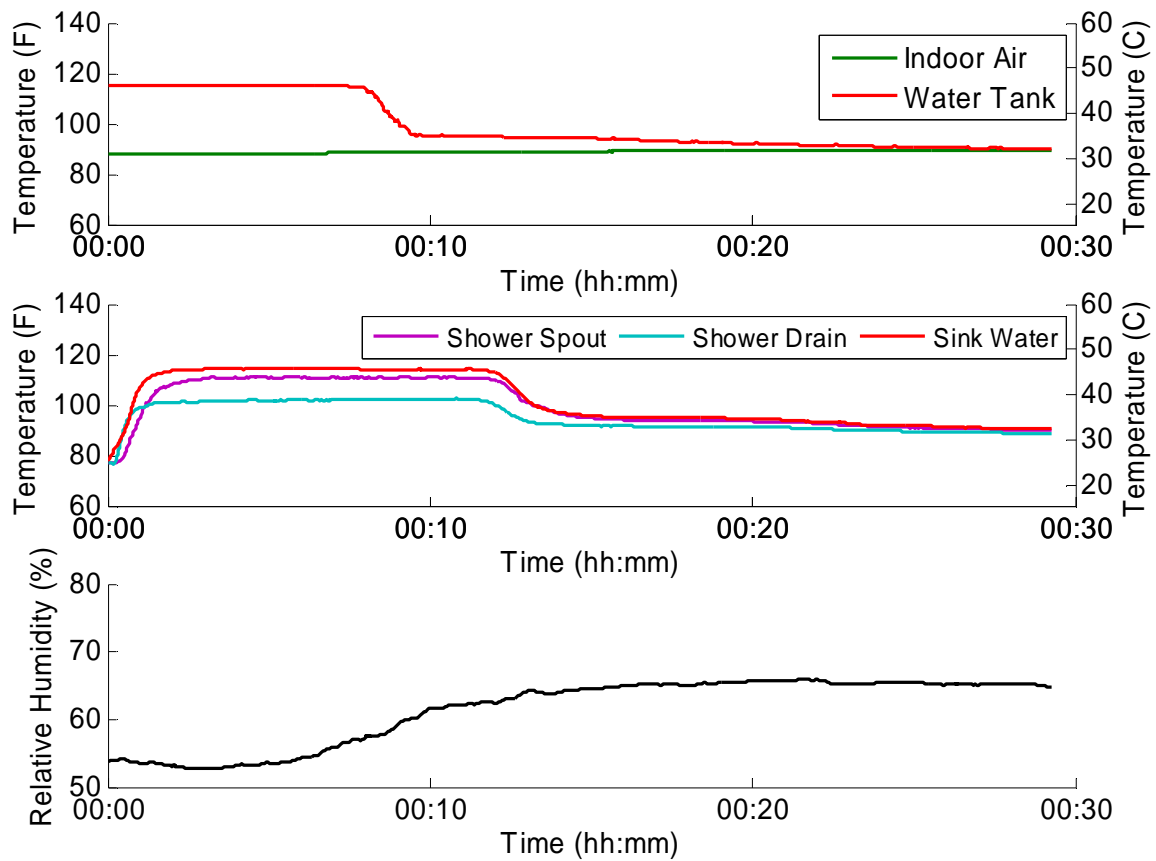


Figure 28. Test 3 house conditions, shower temperature, and water storage temperature.

4.2 STEADY-STATE BEHAVIOR

4.2.1 Experimental Results

At the onset of each test, there was an initial variation of temperature with time for each thermocouple except the one measuring the incoming supply temperature which remained relatively steady over the duration of the test. The temperature variations grew asymptotically to a steady state value which often took several minutes even just to reach 90% of the asymptote value. The flow rates also exhibited obvious transient behavior aside from the pump pulses associated with the inflow rate over the duration of the tests. The flow rates ideally should have

been constant and equal for each test. This, however, was not the case but the average of each flow rate for the different tests can be found in table 2.

Table 2. Mass Flow Rate Comparison.

Average Value	Test 1	Test 2	Test 3
Influent Mass Flow (kg/min)	4.97	6.89	11.31
Effluent Mass Flow (kg/min)	4.88	6.74	11.21
Percent Difference	1.7%	2.2%	0.9%

For each test, the properties of the heat exchanger were calculated for comparison across the test conditions. Among the properties calculated were the log mean temperature difference (ΔT_{lm}), overall heat transfer coefficient times the heat exchanger effective area $(UA)_o$, the effectiveness number (ϵ), and the number of transfer units (NTU). The calculations for these equations can be found in Appendix B. These values were calculated as a function of time for each test. The results of the calculations can be found in figures 29, 30, and 31 for test 1, 2, and 3 respectively.

The $(UA)_o$, ϵ , and NTU for each test was found to be constant over the duration of the test except during the period before the onset of the initial temperature increase induced by the warm waste water flow. The average results of these three heat exchanger properties are tabulated in table 3. The data suggests that as the flow rate through the heat exchanger increases, the effectiveness and number of transfer units decrease while the overall heat transfer coefficient times the heat exchanger effective area increases.

Table 3. Average Heat Exchanger Properties.

Average Value	Test 1	Test 2	Test 3
$T_{Cold,In}$ (°F,°C)	73.0, 22.8	75.9, 24.4	74.1, 23.4
U_oA_o (W/K)	436.6	564.3	754.0
ϵ	0.58	0.55	0.50
NTU	1.32	1.21	0.96

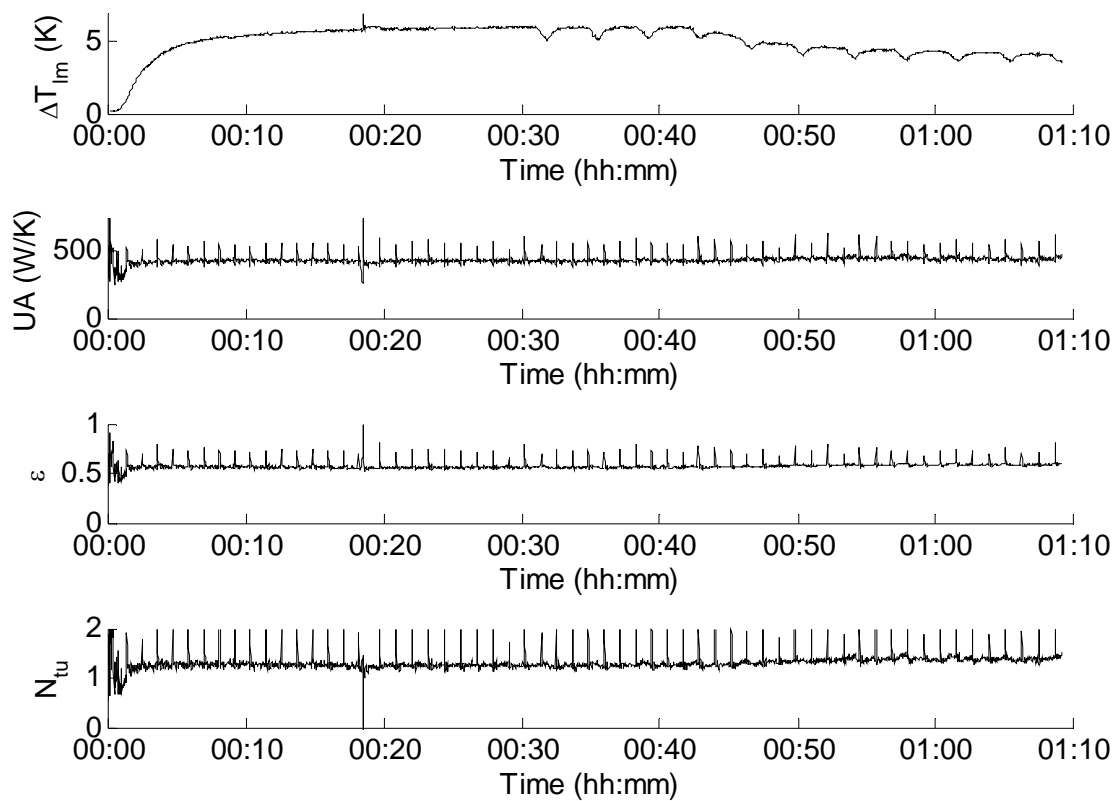


Figure 29. Test 1 heat exchanger performance.

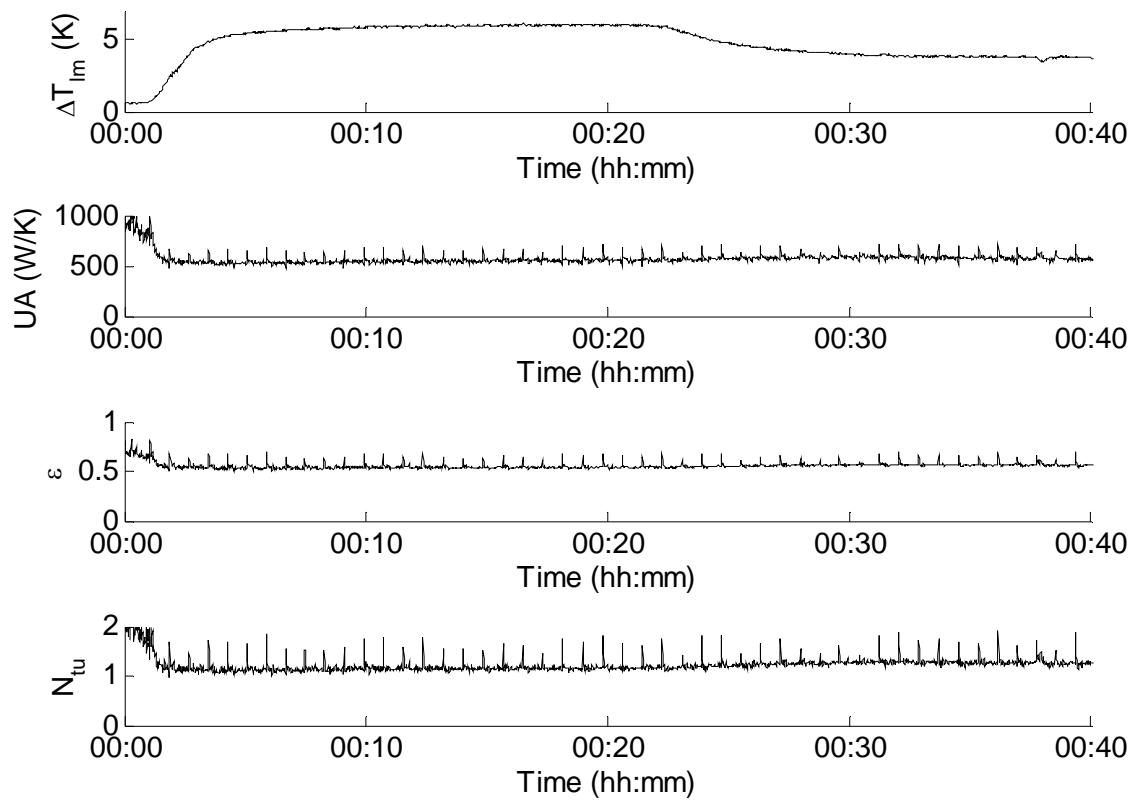


Figure 30. Test 2 heat exchanger performance.

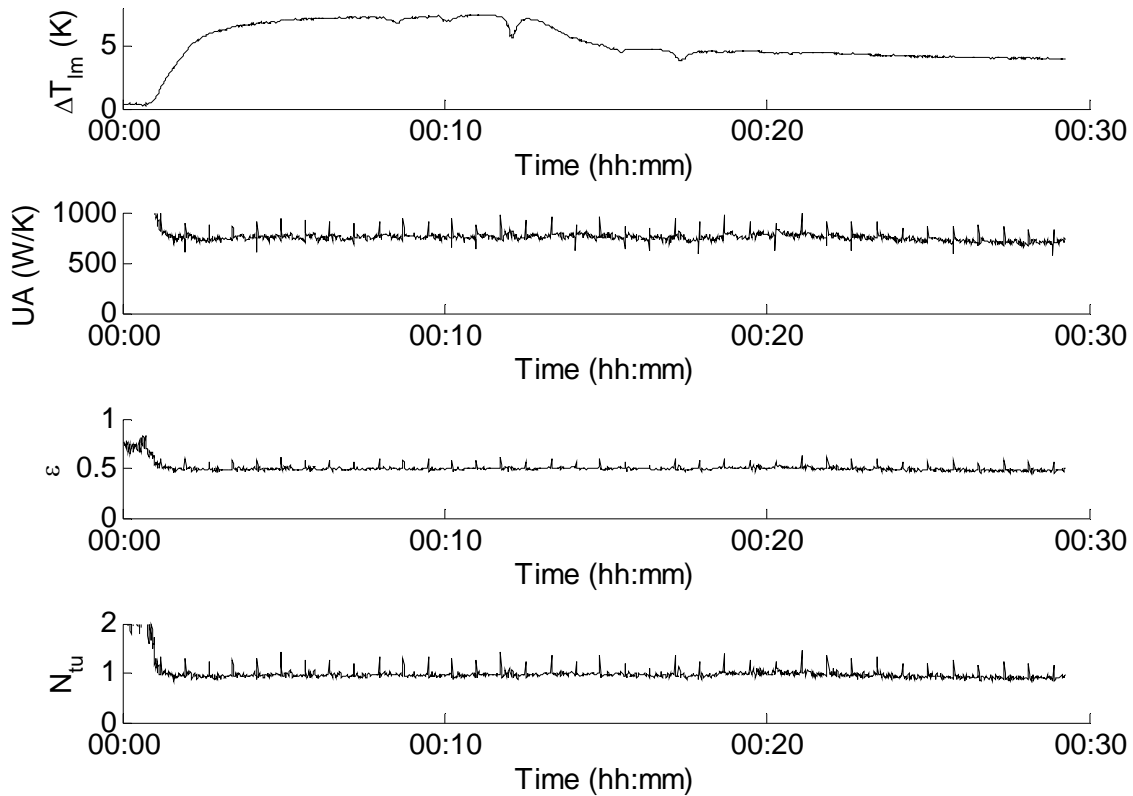


Figure 31. Test 3 heat exchanger performance.

4.2.2 Validation with Manufacturer Data

It is important to compare the results obtained from the experiment with data provided by the manufacturer to validate the accuracy of the experimental results and subsequent calculations. Manufacturer data was acquired for the thermal performance for each of the three test conditions and is tabulated in table 4. A more detailed report of data for each test condition provided by the manufacturer can be found in Appendix C. The hot drain water inlet temperature for the three test conditions is determined from the approximate steady state result from the experiment while the cold inlet temperature and mass flow rates are each averages over the duration of the test. The important parameters provided from that manufacture for

comparison with the experimental data include the heat transfer, overall heat transfer coefficient, and the log mean temperature difference.

Table 4. Thermal Performance Estimated by Manufacturer.

Test 1			Test 2			Test 3		
T _{hi}	97 F	36.1 C	T _{hi}	99 F	37.2 C	T _{hi}	100 F	37.8 C
T _{ci}	73 F	22.8 C	T _{ci}	75.9 F	24.4 C	T _{ci}	74.1 F	23.4 C
m _h	4.88 kg/min		m _h	6.74 kg/min		m _h	11.21 kg/min	
m _c	4.97 kg/min		m _c	6.89 kg/min		m _c	11.31 kg/min	
Q̇	2.7 kW		Q̇	3.4 kW		Q̇	5.9 kW	
U _o	2844 W/m ² -K		U _o	3568 W/m ² -K		U _o	4968 W/m ² -K	
(UA) _o	568.8 W/K		(UA) _o	713.6 W/K		(UA) _o	993.6 W/K	
ΔT _{lm}	5.5 C		ΔT _{lm}	5.6 C		ΔT _{lm}	6.9 C	

For test 1, the manufacturer estimates the heat transfer to be 2.7 kW given the two stream input temperatures and mass flow rates. Comparing this with the experimental results in figure 19, it can be seen that there is minor discrepancy. The experimental results and analysis predict slightly better than 2.5 kW of heat transfer. An important note to make is that the manufacturer data assumes no fouling in the heat exchanger and is therefore a possible explanation for the discrepancy. Another likely explanation for the discrepancy is heat loss to the environment resulting from inadequate insulation of the test apparatus. The manufacturer data for test 2 and 3 conditions predict a heat transfer rate of 3.4 kW and 5.9 kW respectively. Again, looking at the experimental data in figures 23 and 27 it can be seen that the heat transfer of these tests were measured at approximately 3.3 kW and 5.5 kW respectively.

Looking next at the overall heat transfer coefficients provided by the manufacturer for each test condition, there is significant variation with the results calculated from the experimental data. Refer to table 3 for the calculated values of the overall heat transfer coefficients times the effective heat transfer area (436.6 W/K, 564.3 W/K, and 754.0 W/K for tests 1-3 respectively). The manufacturer data specifies each the overall heat transfer coefficient and the effective heat

transfer area separate, but multiplication of the two values for each test condition yield the results in table 4. The manufacturer data is inflated approximately 30%, 26%, and 32% for each test respectively compared to the experiment data.

Finally, looking at the experimentally determined log mean temperature difference for comparison with the information provided by the manufacturer also shows discrepancy. The experimentally determined log mean temperature difference is depicted in figures 29-30 for each test. The values of the log mean temperature difference are approximated at the steady state condition for the experimental results at 6.0 C, 6.0 C, and 7.3 C for tests 1-3 respectively. The experimental results are approximately 9%, 7%, and 6% inflated from the manufacturer data.

4.2.3 Validation with a Theoretical Model

Another method that can be used to validate the experimental data is to use a theoretical model of the heat exchanger to predict thermal performance. This validation process began by modeling the plate heat exchanger (PHE) in Engineering Equation Solver (EES). For this particular application of a heat exchanger, it was chosen to use the NTU-effectiveness method to determine the heat exchanger capacity. This method was chosen because the effluent and influent stream inlet temperatures to the heat exchanger are dictated by the house water drain water temperature and the temperature of the cold supply water respectively and there is no explicit constraint on the outlet temperatures. From there, the heat exchanger capacity is dictated by the geometry of the heat exchanger and the flow rates through the heat exchanger. The geometry is used to determine the overall heat transfer coefficient which is then used to find NTU, effectiveness, and ultimately the heat recovery rate.

The overall heat transfer coefficient (U_o) between the two flow streams is defined by equation 3 which represents the resistive network that accounts for convective heat transfer in

each flow as well as the conduction through the separating wall with thickness (t). In equation 3, k_w is the thermal conductivity of the separating wall material while h_c and h_h are the convective heat transfer coefficients of the cold flow and hot flow respectively.

$$\frac{1}{U_o} = \frac{1}{h_c} + \frac{t}{k_w} + \frac{1}{h_h} \quad (3)$$

$$h = \frac{Nu \cdot k}{D_h} \quad (4)$$

The convective heat transfer coefficient is found using equation 4 and requires the knowledge of the Nusselt number for each flow stream. There are several publications that exist for the purpose of characterizing the Nusselt number and friction factors in chevron-style plate heat exchangers based on geometric features. The common purpose of these studies is to determine how geometric features of the PHE affect the thermal and hydraulic performance of the unit. The geometric features commonly found to affect the PHE performance include chevron angle (β), corrugation depth (b), corrugation pitch (Λ), and area enlargement ratio (Φ). Figure 32 shows how each parameter is commonly defined with the exception of the area enlargement ratio which is the ratio of actual heat transfer area to the projected area.

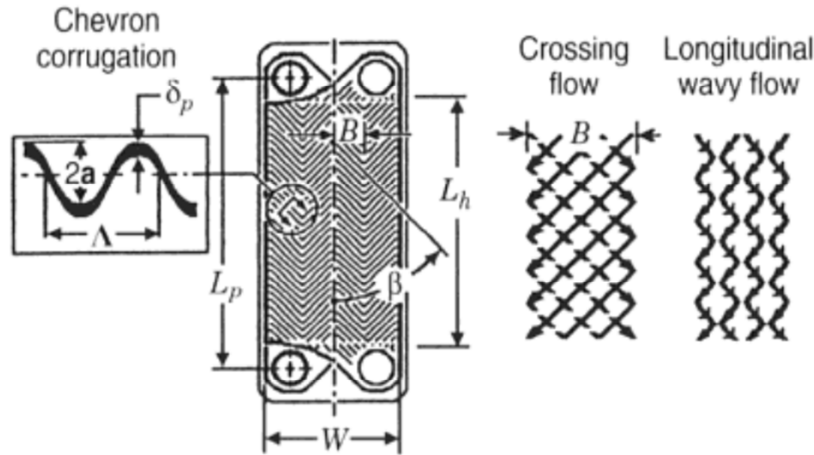


Figure 32. Diagram of chevron-style PHE (Martin, 1996).

This latter factor is a result of the corrugation pattern that is usually stamped into the plates and increases actual surface area of the plate while maintaining a relatively constant projected area dictated by the original length and width of the plate.

Corrugation depth and pitch are related to the area enlargement ratio and therefore are commonly represented by this single parameter. Equations 5 and 6 offer an approximation from Martin (1996) for the relation between these parameters. It should be noted that the corrugation depth is related to the corrugation amplitude by equation 7.

$$\Phi \approx \frac{1}{6} \left(1 + \sqrt{1 + \chi^2} + 4 \sqrt{1 + \frac{\chi^2}{2}} \right) \quad (5)$$

$$\chi = \frac{2\pi a}{\Lambda} = \frac{\pi b}{\Lambda} \quad (6)$$

$$b = 2a \quad (7)$$

Some studies like the one completed by Muley and Manglik (1999) that investigate the effect of PHE geometry on thermal and hydraulic performance have developed relations for the Nusselt number by fitting expressions similar to the Dittus-Boelter or Seider-Tate correlation to experimental data. These expressions often include terms dependent on aforementioned geometry factors and are typically fit for a certain Reynolds number range. However, another study by Martin (1996) developed a semi-theoretical expression for Nusselt number and friction factor based on an extension of L  v  que theory into the turbulent regime. The result is a semi-theoretical expression because the form is fit to experimental data. The correlation developed by Martin (1996) was suggested to yield valid results over a large corrugation angle range (between 10 and 80 degrees) by Wang (2003) which is larger than the suggest range for the expressions suggested by Muley and Manglik (1999) which are valid between 30 and 60 degrees. A study by

Claesson (2005) found that the expression developed by Martin (1996) better predicts the manufacturer published data for a specific heat exchanger compared to the expressions suggested by Muley and Manglik (1999). Therefore, the expressions used to estimate the Nusselt number (Nu) and Moody friction factor (f) are given by equations 8 and 9-11 respectively.

$$Nu = 0.122 \cdot Pr^{1/3} \left(\frac{\mu}{\mu_w} \right)^{1/6} \left[f \cdot Re_h^2 \cdot \sin 2\beta \right]^{0.374} \quad (8)$$

$$\frac{1}{\sqrt{f}} = \frac{\cos \beta}{\sqrt{0.18 \cdot \tan \beta + 0.36 \cdot \sin \beta + \frac{f_0}{\cos \beta}}} + \frac{1 - \cos \beta}{\sqrt{3.8 \cdot f_1}} \quad (9)$$

$$f_0 = \frac{64}{Re_h} \quad \text{if } Re_h < 2000 \quad (10)$$

$$f_0 = (1.8 \cdot \log_{10} Re_h - 1.5)^{-2} \quad \text{if } Re_h \geq 2000$$

$$f_1 = \frac{597}{Re_h} + 3.85 \quad \text{if } Re_h < 2000 \quad (11)$$

$$f_1 = \frac{39}{Re_h^{0.289}} \quad \text{if } Re_h \geq 2000$$

The Nusselt number given by equation 8 is a function of the corrugation angle and Moody friction factor, but also the Prandtl number (Pr) of the flow, the dynamic viscosity of the bulk flow (μ), the dynamic viscosity of the flow at the wall (μ_w), and the Reynolds number based on the hydraulic diameter (Re_h). The Reynolds number based on the hydraulic diameter is defined by equation 12 where the hydraulic diameter (D_h) is defined by equation 13. The Reynolds number is also a function of the density of the stream (ρ), the velocity (u), and the corrugation angle. The flow velocity is approximated using equation 14 where Q is the volumetric flow rate, w is the PHE width, and N is the number of channels for the flow stream.

$$Re_h = \frac{\rho u \cos \beta}{\mu} \cdot D_h \quad (12)$$

$$D_h = \frac{4a}{\Phi} = \frac{2b}{\Phi} \quad (13)$$

$$u = \frac{Q}{N \cdot w \cdot b} \quad (14)$$

After solving the overall heat transfer rate, the number of transfer units (NTU) can be calculated using equation 15. A_o is the projected area of a single plate defined by equation 16 where L_p is the length of a plate from port center to center (see figure 2). Also used to define NTU is the number of plates (N_{plates}) and C_{min} , the minimum heat capacity of either flow.

$$NTU = \frac{(UA)_o \cdot N_{plates}}{C_{min}} \quad (15)$$

$$A_o = w \cdot L_p \quad (16)$$

With the calculated NTU, effectiveness (ε) can be determined using the ε -NTU relation for a counter flow heat exchanger, equations 17 and 18.

$$\varepsilon = \frac{1 - \exp[-NTU(1 - C_r)]}{1 - C_r \cdot \exp[-NTU(1 - C_r)]} \quad (17)$$

$$C_r = \frac{C_{min}}{C_{max}} \quad (18)$$

Finally, the heat recovery rate can be calculated using equations 19 and 20 where q_{max} is the maximum possible heat transfer rate, T_{hi} is the drain water inlet temperature and T_{ci} is the cold supply water inlet temperature.

$$q = \varepsilon \cdot q_{max} \quad (19)$$

$$q_{max} = C_{min}(T_{hi} - T_{ci}) \quad (20)$$

After solving the heat recovery rate, the temperature of the exiting flows can be determined using equation 1 which then enables the calculation of the log mean temperature

difference in the heat exchanger. The expression for the log mean temperature difference is a function of the inlet and outlet temperatures of both flow streams and is given as equation 32. This parameter can also be related to the heat transfer and overall heat transfer coefficient by the expression given in equation 33.

$$\Delta T_{lm} = \frac{(T_{H,in} - T_{C,out}) - (T_{H,out} - T_{C,in})}{\ln \left[\frac{(T_{H,in} - T_{C,out})}{(T_{H,out} - T_{C,in})} \right]} \quad (32)$$

$$(UA)_o = \frac{\dot{Q}}{\Delta T_{lm}} \quad (33)$$

The geometry of the PHE used in this study is listed in table 5 while additional specifications and product details can be found in Appendix C. The geometry data in table 5 was acquired by measuring a sample plate provided by the manufacturer which can be seen in figure 33. Each dimension listed in table 5 is a hand measurement except for the area enlargement ratio which is calculated from the corrugation depth and pitch using equations 5 and 6. Using this geometry data, the thermal performance data was predicted for each of the three test conditions. The results of the model prediction are listed in table 6.

Table 5. Flat Plate Heat Exchanger Plate Measurements.

Plate Thickness , t	0.43 mm	0.017 in
Corrugation Pitch, Λ	7.3 mm	0.287 in
Corrugation Depth, b	2.1 mm	0.083 in
Corrugation Angle, β	25 deg	
Area Enlargement Ratio, Φ	1.18	

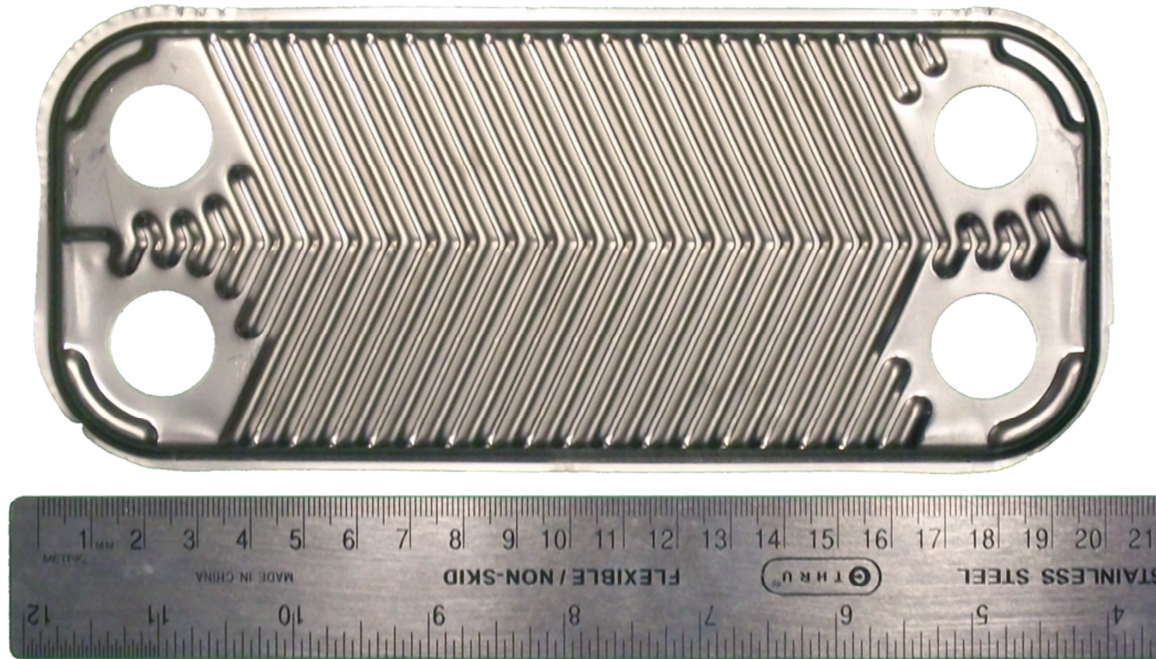


Figure 33. Diagram of plate in flat plate heat exchanger.

Table 6. Thermal Performance Estimated by Theoretical Model.

Test 1			Test 2			Test 3		
T _{hi}	97 F	36.1 C	T _{hi}	99 F	37.2 C	T _{hi}	100 F	37.8 C
T _{ci}	73 F	22.8 C	T _{ci}	75.9 F	24.4 C	T _{ci}	74.1 F	23.4 C
m _h	4.88 kg/min		m _h	6.74 kg/min		m _h	11.21 kg/min	
m _c	4.97 kg/min		m _c	6.89 kg/min		m _c	11.31 kg/min	
Q̇	1.46 kW		Q̇	1.77 kW		Q̇	2.82 kW	
U _o	909 W/m ² -K		U _o	1099 W/m ² -K		U _o	1476 W/m ² -K	
(UA) _o	160.7 W/K		(UA) _o	194.4 W/K		(UA) _o	261 W/K	
ΔT _{lm}	9.06 C		ΔT _{lm}	9.08 C		ΔT _{lm}	10.81 C	

The results of the model yield similar trends to the experimental and manufacturer data; however the predicted magnitude of heat transfer is significantly reduced. The model predicts 1.46 kW, 1.77 kW, and 2.82 kW of heat recovery for tests 1-3 while experimental results were approximately 2.5 kW, 3.3 kW, and 5.5 kW respectively. Therefore the model underestimated the actual heat transfer by 41.6%, 46.4%, and 48.7% for each test respectively. The model, as described, has proved incapable of accurately predicting the thermal performance of the actual

heat exchanger and requires improvement before it can be used as a design tool to optimize heat exchanger geometry for the given flow conditions of this application.

4.3 ENERGY RECOVERY POTENTIAL

The energy recuperated by the heat recovery system was calculated as a function of time for each test. Equation 21 was used to quantify the percentage of energy that can be recovered if the heat capacity of water is assumed to be constant.

$$\%E_{recovery} = \left[\frac{(T_{C,out} - T_{C,in})}{(T_{Fixture\ Water} - T_{C,in})} \right] \times 100\% \quad (21)$$

In equation 21, the temperature of the shower spout appears to influence the energy recovery potential for the hot water heating system, however, it should be noted that as the shower spout temperature decreases, so will the temperature of the waste water feeding into the heat recovery system. Consequently, this will reduce the outlet temperature of the cold side of the heat exchanger. Results published by ORNL (2005) indicate that shower temperature does not have a significant impact on the energy savings for the GFX. The energy recovery can be assumed equal to energy savings for the hot water heating system if temperature losses in pipes are neglected. Figure 34 shows the percent energy recovery for the first test.

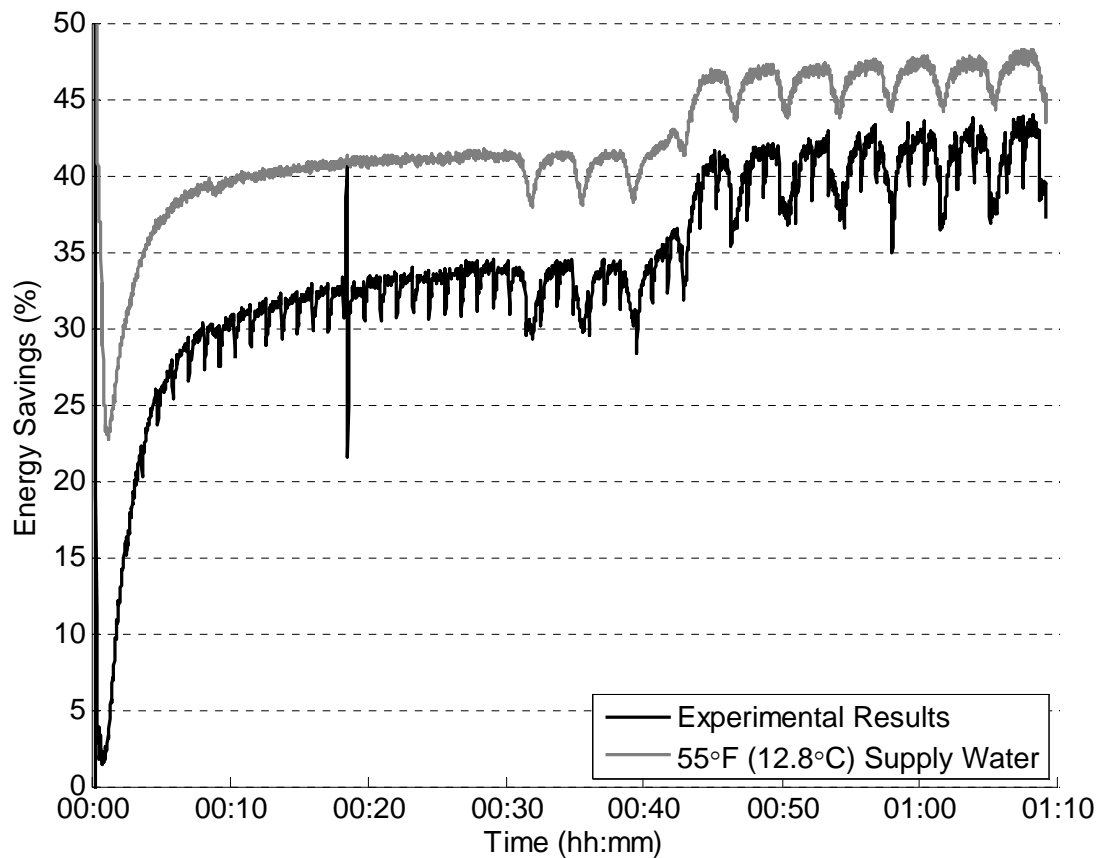


Figure 34. Test 1 time dependent energy savings potential.

Figure 34 also shows the predicted percent energy recovery of the system for cold supply water that is at a lower temperature than the supply utilized for the given experiment. A prediction for 55 °F (12.8 °C) supply water is depicted in figure 34 and was chosen by suggestion from Chinery (2004) that a decent approximation for the annual average water main temperature is the average annual ambient air temperature for the geographic location. According to TMY3 data sets produced by the National Renewable Energy Laboratory, the average ambient air temperature in central Illinois ranges between 50 and 55 °F (10 and 37.8 °C) depending on the specific location of interest.

The predicted results were obtained by solving equation 2 for the cold side outlet temperature of the heat exchanger. By assuming balanced flow and a constant and equal specific heat capacity for both inlet and outlet flow through the heat exchanger, equation 22 results.

$$T_{C,out} = \varepsilon(T_{H,in} - T_{C,in}) + T_{C,in} \quad (22)$$

After solving the cold side outlet temperature with equation 22 assuming that ε is the same as the calculated value from experimental data, then equation 21 was used to find the predicted percent energy savings for the associated water supply inlet temperature and fixture water temperature. Figure 35 and 36 show the results of the energy savings analysis for test 2 and 3 respectively.

The first test with an average incoming water supply temperature of 73.0 °F (22.8 °C) had an energy recovery rate which approached 34% for the experiment. If the incoming water supply had been 55 °F (12.8 °C), then the recovery would have been approximately 41%. For the second test that had an average incoming water supply temperature of 75.9 °F (24.4 °C), had an energy recovery that approached 37%, and again if the water inlet temperature had been 55 °F (12.8 °C), the energy recovery would have improved to approximately 44%. For the third test with average incoming water temperature of 74.1 °F (23.4 °C), was approximately 33% with an improved recovery of approximately 38% for 55 °F (12.8 °C) average inlet supply temperature. These approximations are tabulated in table 7. It should be noted that for test 2 and 3, the fixture temperature is assumed to be the temperature of the sink which is slightly higher than the shower temperature during the tests. This fact causes the energy savings prediction to be more conservative than if the fixture temperature was a mass flow rate based average for the two separate flow temperatures.

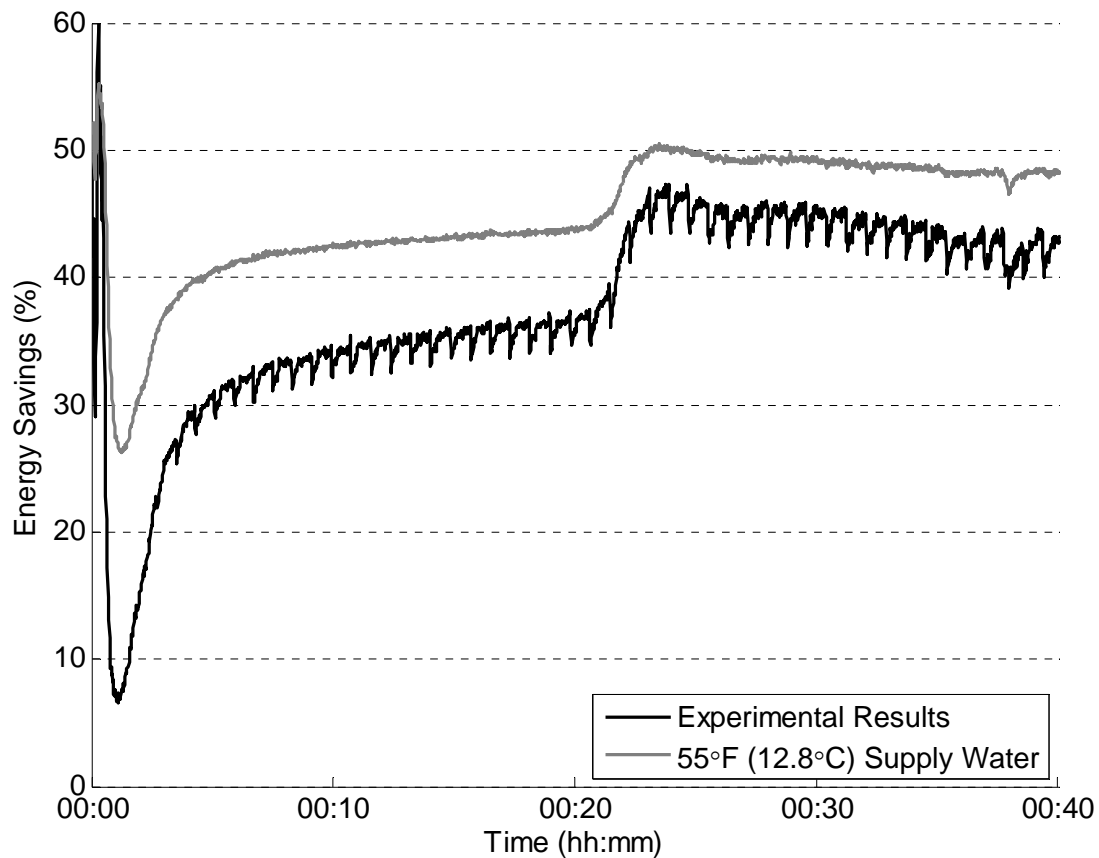


Figure 35. Test 2 time dependent energy savings potential.

All recovery potential is evaluated before the thermocline in the tank is breeched. Once the thermocline is breeched, the temperature of the shower decreases while stored thermal energy in the pipes and drains of the drainage system continue to provide warm waste water to the heat exchanger therefore maintaining a high cold side outlet temp and an inflated recovery potential value. This data is not erroneous, but merely irrelevant as personal comfort is compromised once the thermocline is breeched and a shower will typically be terminated at that time, therefore, making this operating condition significantly less frequent.

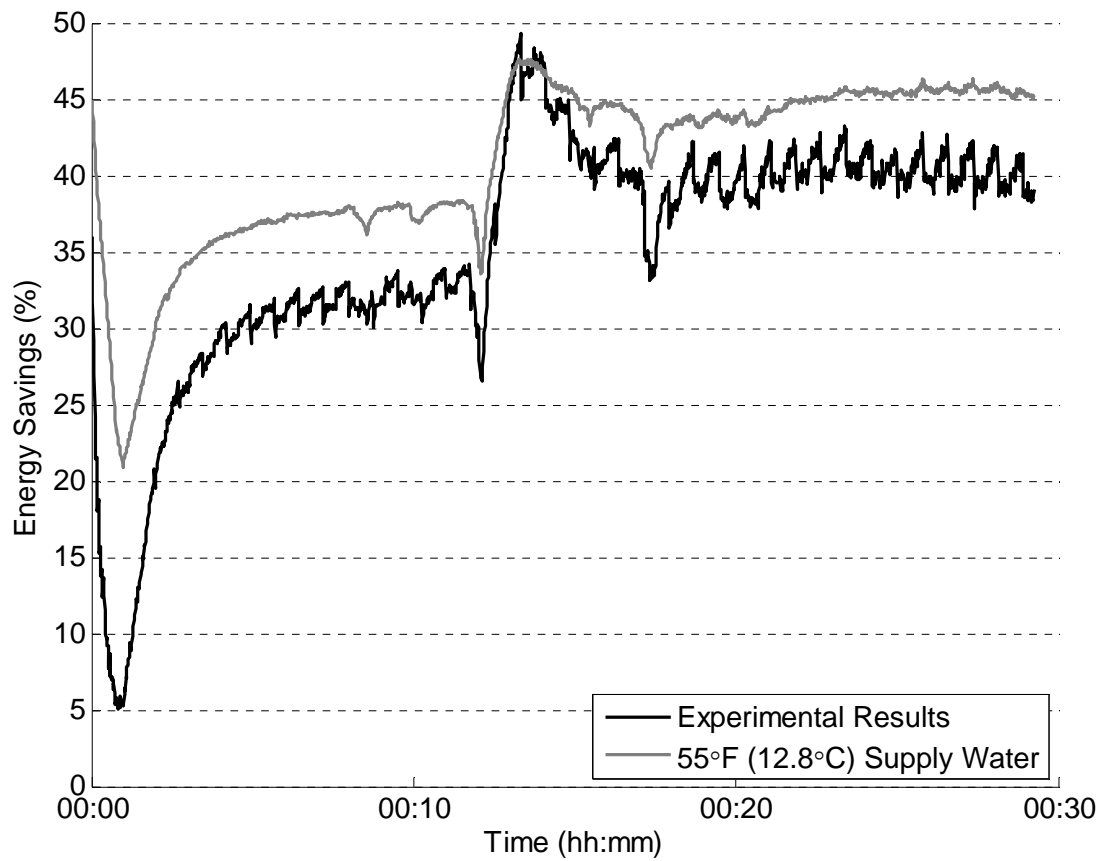


Figure 36. Test 3 time dependent energy savings potential.

Table 7. Heat Recovery Results.

	Test 1	Test 2	Test 3
Average supply mass flow rate (g/s)	82.8	114.8	188.5
Average supply water temperature (°F,°C)	73.0, 22.8	75.9, 24.4	74.1, 23.4
Average shower temperature (°F,°C)	114.3, 45.7	109.3, 42.9	110.9, 43.8
Experimental heat recovery	34%	37%	33%
Predicted heat recovery (for 55°F, 12.8°C supply water)	41%	44%	38%

CHAPTER 5: CONCLUSIONS

The goal of this study was to investigate heat recovery from domestic waste water and to help identify the predominant influential factors to system design and performance. The first objective was to identify residential waste water heat recovery system designs and characterize the potential benefits and limitations of each. There were several systems identified in scientific literature that investigated the use of building waste water heat recovery. It was discovered that several system design options and heat recovery methods were previously researched including a non-regenerative heat pipe system used for preheating incoming supply water. This system was found to have an effectiveness of approximately 90% for recovering usable heat from a waste water flow (Behrmann, 1982). Another system discovered was a thermosyphon system that is regenerative system used to preheat incoming ventilation air for a home. This study found that up to 30% of the energy used to heat the water could be recovered, but its usefulness was limited to the heating season (Kalema et al., 1985). There were several waste water heat recovery systems identified that utilized various heat exchanger configurations. The study performed by Smith (1975) utilized a regenerative tank in tank heat exchanger to save 31.7% on water heating energy in a winter season by preheating incoming supply water. It was estimated that the system could maintain 30% water heating energy savings throughout the entire year. Wong et al. (2010) investigated the use of a non-regenerative tube in tube heat exchanger for distributed energy recovery on the drain line of showers which would preheat cold water supplying the instantaneous water heater servicing the particular shower. The results of the study found that between 4% and 15% of energy savings on hot water heating could be obtained with this heat exchanger. A third heat exchanger based waste water heat recovery system identified was the non-regenerative GFX system used to preheat incoming supply water that was validated by

ORNL (2005) to be capable of reducing water heating energy consumption by 40%. Another type of heat recovery system identified is one that utilizes a heat pump and heat exchanger to recover waste heat from a public shower facility to preheat the incoming supply water. This non-regenerative system is expected to reduce electricity consumption by 44.5% (Liu et al., 2010). The knowledge and insight developed in the aforementioned studies is the base to the experiment design choices made for the system further investigated in this study.

The second objective of this study was to design an experiment to validate the energy savings potential of waste water heat recovery with data published in scientific literature. For this study, a non-regenerative, centralized, counter flow, heat exchanger based waste water heat recovery system was developed and used to preheat incoming supply water for a test home to characterize energy savings potential while examining the effects of varying flow rates in a balanced flow configuration. Three tests with varying flow rate were conducted, running non-accumulating hot water fixtures at max temperature, and each indicated that a 34-37% recovery in hot water heating energy is practical at steady-state conditions. This is lower than the similar heat recovery system published in literature by ORNL (2005) that demonstrated the GFX heat recovery device permits up to 40% savings in water heating energy. The final objective of the study is meant to further explain these variances.

The final objective of this study was to provide data to help identify essential parameters for inclusion in waste water heat recovery prediction models. The study by ORNL (2005) determined for waste water heat recovery systems that preheat the incoming supply water to a residence in a non-regenerative scheme with a heat exchanger are less effective toward energy savings when installed in unbalanced flow configurations compared to a balanced flow configuration. It is therefore important to know the capacity ratio between the streams as this

governs the effectiveness of a heat exchanger. For the case where the heat capacity of the waste water discharge stream exceeds that of the incoming heat sink stream of the heat exchanger, the effectiveness decreases. The alternative cases where the heat capacity of cold sink exceeds that of the discharge flow, the heat exchanger could prove more effective at recovering waste heat, but this condition does not occur for non-regenerative systems. The study by ORNL (2005) also indicated a relative independence of the end-use hot water fixture temperature on energy savings for balanced flow conditions. This cannot be confirmed with the data from the present study due to the variability of the average incoming supply water temperature characteristic to above ground water storage of the test house installation. It is, however, predicted that the average inlet supply water temperature does affect the heat recovery potential for the given experiment. In each test, the predicted energy recovery increases 5% to 7%. The varying flow rates between each test have little discernable influence on the heat recovery of the waste water. Given a higher flow rate, the heat transfer is higher; however, the calculated heat exchanger effectiveness only decreases from 0.58 for the lowest flow case, to 0.50 for the highest flow case. Even this variance is unexpected, and a source for improvement in future experiments along with establishing a water supply source with a more constant temperature.

Overall, the potential for waste water heat recovery in a residential application has been characterized for a non-regenerative, centralized, flat plate heat exchanger based heat recovery system used to preheat incoming supply water in a balanced flow installation. The amount of recoverable heat has proven substantial with up to 37% recovery. Waste water heat recovery is an area deserving of continued research given the potential energy savings benefit that can be attained, and energy conservation is a responsibility that everyone shares.

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APPENDIX A

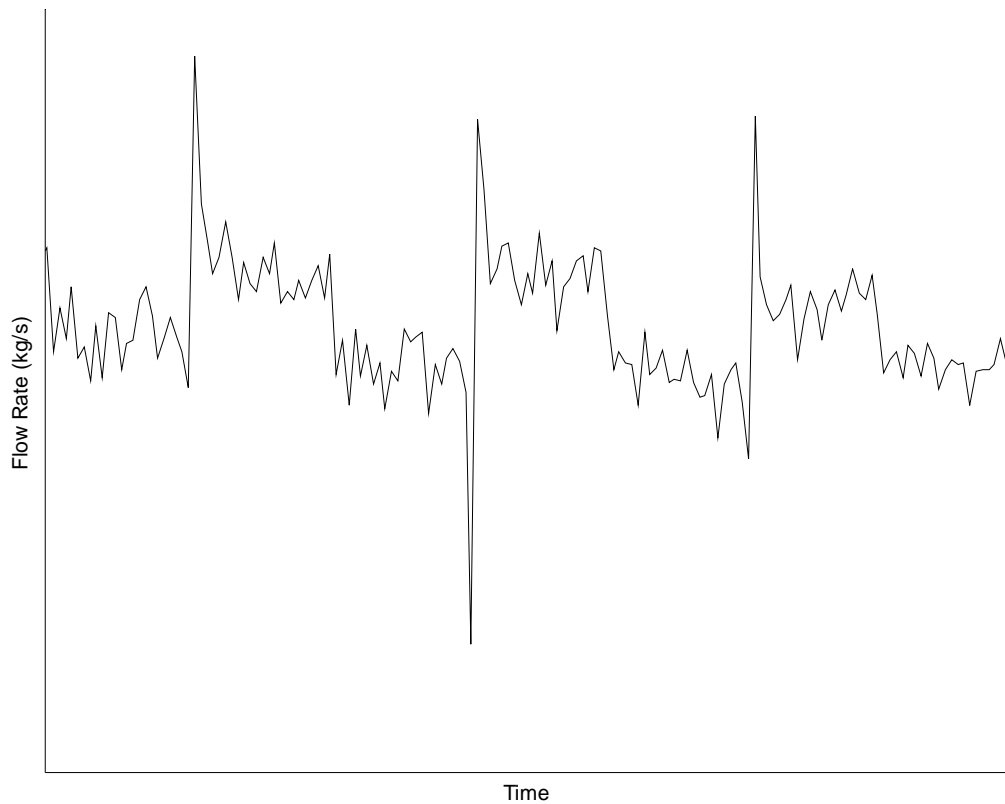


Figure 37. Effect of unsteady supply pressure on supply flow rate.

The supply water for the test house is supplied by a jet pump with limited pressure storage capacity. The jet pump cycled on and off as the water pressure dropped below a low pressure threshold and when it reached above the high pressure threshold. It is assumed that this cycling effect caused a variation in the supply line pressure, therefore altering the flow rate to the house. The cycling was more frequent for higher flow rates. Figure 37 displays the effect of the unsteady pressure on the flow rate even with a pressure regulator on the supply line to help mitigate the effects of this pump cycling.

APPENDIX B

Energy Balance for a Heat Exchanger

Starting with the conservation of energy for one dimensional flow through a control volume (equation 23), and assuming the control volume around the entire heat exchanger is at steady-state, then $\frac{dE_{cv}}{dt} = 0$.

$$\frac{dE_{cv}}{dt} = \dot{Q}_{cv} - \dot{W}_{cv} + \sum_i \dot{m}_i \left(h_i + \frac{V_i^2}{2} + gz_i \right) - \sum_e \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gz_e \right) \quad (23)$$

Also, because only flow work is present in the control volume, $\dot{W}_{cv} = 0$. Kinetic and potential energy contributions are ignored therefore, $\sum_i \dot{m}_i \left(\frac{V_i^2}{2} + gz_i \right) - \sum_e \dot{m}_e \left(\frac{V_e^2}{2} + gz_e \right) = 0$. This allows equation 23 to reduce to equation 24.

$$\sum_e \dot{m}_e (h_e) - \sum_i \dot{m}_i (h_i) = \dot{Q}_{cv} \quad (24)$$

For a heat exchanger with an inlet and outlet for two flow streams, one hot and one cold, equation 24 can be represented according to equation 25.

$$(\dot{m}h)_{C,in} + (\dot{m}h)_{H,in} + \dot{Q}_{lost} = (\dot{m}h)_{C,out} + (\dot{m}h)_{H,out} \quad (25)$$

By assuming $\dot{m}_{C,in} = \dot{m}_{C,out}$ and $\dot{m}_{H,in} = \dot{m}_{H,out}$, equation 25 can be reduced to equation 6.

$$\dot{m}_C(h_{C,out} - h_{C,in}) = \dot{m}_H(h_{H,in} - h_{H,out}) + \dot{Q}_{lost} \quad (26)$$

Using the definition of enthalpy according to equation 27, and finding the change in enthalpy between two states (equation 28), the definition for change in internal energy (equation 29) can be substituted into equation 28. Also assuming incompressibility, $c_v = c_p = c$ and $v_2 = v_1 = v$, equation 28 can be reduced to equation 30.

$$h = u + pv \quad (27)$$

$$h_2 - h_1 = u_2 - u_1 + (pv)_2 - (pv)_1 \quad (28)$$

$$u_2 - u_1 = \int_{T_1}^{T_2} c_V(T) dT \quad (29)$$

$$h_2 - h_1 = c(T_2 - T_1) + v(p_2 - p_1) \quad (30)$$

Finally, assuming $p_2 \approx p_1$, equation 30 can be substituted into equation 6, therefore reducing the energy balance to equation 31 where \dot{Q}_{lost} is negative to account for heat lost to the environment and positive if heat is gained from the environment.

$$[\dot{m}c(T_{\text{out}} - T_{\text{in}})]_C = [\dot{m}c(T_{\text{in}} - T_{\text{out}})]_H + \dot{Q}_{\text{lost}} \quad (31)$$

APPENDIX C

Table 8. Experimental Apparatus Parts Description.

Part	Quantity	Model Number/Description
Flat Plate Heat Exchanger	1	FlatPlate FP3x8-14 (3/4" MPT)-MMC
Diaphragm Pump	1	SHURflo 2088-594-144BX
T-type thermocouple	7	TMQSS-125G-6
Flow meter	2	FTB-4607 Hall Effect FlowMeter

Table 9. Flat Plate Heat Exchanger Physical Properties.

No. of Plates	14	Braze Material	Copper
Width	3.3 in, 83.82 mm	Maximum Working Temperature	350 F, 176.67 C
Length	7.8 in, 198.12mm	Minimum Working Temperature	-320 F, -195.56 C
Depth	1.6 in, 40.64mm	Maximum Working Pressure	450 psig, 3102.6 kPa
Connections	3/4" MPT		
Weight	3.2 lbs, 1.45 kg		
Plate Material	316L Stainless Steel		

Table 10. Heat Exchanger Manufacturer Data for Test 1 Conditions.

Model: FP3X8-14 (3/4" MPT)			
Load (kW)	2.7	Nominal surface (m ²)	0.2
Log mean temp. diff. (°C)	5.5	Dimensions	83W x 197H x 40D
Overall HTC (W/m ² K)	2,844	Plate construction	Single wall
Oversurface percent	0.0	Net weight (kg)	1.4
Model size	3x8		
Design Conditions	Side A - Liquid	Side B - Liquid	
Fluid type	Water	Water	
Fluid mass flow rate (kg/min)	4.9	5.0	
Entering fluid temp. (°C)	36.1	22.8	
Leaving fluid temp. (°C)	28.2	30.6	
Fluid flow rate (L/min)	4.9	5.0	
Fluid fouling factor (m ² ·K/W)	0.000000	0.000000	
Model Parameters			
Number of channels	6	7	
Velocity (m/s)	0.09	0.08	
Pressure drop (kPa)	1.3	1.0	
Heat transfer coef. (W/m ² K)	6,045	5,370	
Internal volume (L)	0.200	0.233	

Table 11. Heat Exchanger Manufacturer Data for Test 2 Conditions.

Model: FP3X8-14 (3/4" MPT)			
Load (kW)	3.4	Nominal surface (m²)	0.2
Log mean temp. diff. (°C)	5.6	Dimensions	83W x 197H x 40D
Overall HTC (W/m²K)	3,568	Plate construction	Single wall
Oversurface percent	0.0	Net weight (kg)	1.4
Model size	3x8		
Design Conditions	Side A - Liquid	Side B - Liquid	
Fluid type	Water	Water	
Fluid mass flow rate (kg/min)	6.7	6.9	
Entering fluid temp. (°C)	37.2	24.4	
Leaving fluid temp. (°C)	29.9	31.6	
Fluid flow rate (L/min)	6.8	6.9	
Fluid fouling factor (m²·K/W)	0.000000	0.000000	
Model Parameters			
Number of channels	6	7	
Velocity (m/s)	0.13	0.11	
Pressure drop (kPa)	2.4	1.9	
Heat transfer coef. (W/m²K)	7,575	6,746	
Internal volume (L)	0.200	0.233	

Table 12. Heat Exchanger Manufacturer Data for Test 3 Conditions.

Model: FP3X8-14 (3/4" MPT)			
Load (kW)	5.9	Nominal surface (m ²)	0.2
Log mean temp. diff. (°C)	6.9	Dimensions	83W x 197H x 40D
Overall HTC (W/m ² K)	4,968	Plate construction	Single wall
Oversurface percent	0.0	Net weight (kg)	1.4
Model size	3x8		
Design Conditions	Side A - Liquid	Side B - Liquid	
Fluid type	Water	Water	
Fluid mass flow rate (kg/min)	11.2	11.3	
Entering fluid temp. (°C)	37.8	23.4	
Leaving fluid temp. (°C)	30.2	30.9	
Fluid flow rate (L/min)	11.3	11.4	
Fluid fouling factor (m ² ·K/W)	0.000000	0.000000	
Model Parameters			
Number of channels	6	7	
Velocity (m/s)	0.22	0.19	
Pressure drop (kPa)	6.2	4.9	
Heat transfer coef. (W/m ² K)	10,631	9,324	
Internal volume (L)	0.200	0.233	